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NRL Report 4789

DESIGN OF SHOCK- AND VIBRATION-RESISTANT ELECTRONIC EQUIPMENT FOR SHIPBOARD USE

Harold M. Forkois and Kenneth E. Woodward

Shock and Vibration Branch
Mechanics Division

September 19, 1956

NAVAL RESEARCH LABORATORY
Washington, D.C.

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2. Electronic equipment - Vibration
3. Electronic equipment - Design

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CONTENTS

ABSTRACT

The structural design requirements of shock- and vibration-resistant shipboard electronic equipment are stiffness and lightness, which imply high structural natural frequencies. These two fundamental design characteristics are advanced principally on the basis of practical solutions of problems involved in evaluations of electronic equipment at NRL, with respect to their ability to withstand shock and vibration phenomena. Compliance with these practical guides will eliminate most of the avoidable difficulties that repeatedly occur in evaluation procedures, and will contribute to the development of reliable equipment.

PROBLEM STATUS

This is a final report on one phase of the problem; work on the general problem is continuing.

AUTHORIZATION

NRL Problem F02-05
BuShips Project NS 711-105

*Manuscript submitted June 21, 1956

DESIGN OF SHOCK- AND VIBRATION-RESISTANT ELECTRONIC EQUIPMENT FOR SHIPBOARD USE

Harold M. Forkois and Kenneth E. Woodward

INTRODUCTION

Increases in the destructive power of weapons, and the increased utilization and complexity of electronic apparatus, has brought about a need for the highest degree of reliability in the performance of all equipment. It may appear trite to say that equipment reliability must have its inception in the initial design stages, but it is an unfortunate fact that many design deficiences which are uncovered by shock and vibration tests could, in the great majority of cases, have been avoided quite easily by proper consideration of the design.

It is the purpose of this report to provide design engineers of electronic equipment, particularly for shipboard equipment, with practical information relating to what constitutes good and bad features of mechanical design. The mechanical design considerations presented are based on evaluations at the Naval Research Laboratory involving many hundreds of equipments. The procedures involved in conducting these shock and vibration evaluations are developmental in nature. The equipments are operated and monitored so that an adequate determination of performance will result. If an assembly ceases to operate, functions improperly, or gives spurious information, the test is stopped and the causes of improper operation are determined. Corrective measures are then taken to restore operation and to eliminate faults before the test is continued. These corrective measures, performed within the limits of time and personnel available, often constitute a complete redevelopment of structural parts of the equipments. When the test is completed, recommendations for improvement (based on corrective measures incorporated during the testing procedure) are made in a written report. Approval of equipment is contingent upon compliance with these recommendations. A previous publication (1) summarized actual damages that occurred during shock and vibration tests. This report shows how many of these damages could have been avoided.

DEFINITIONS

Vibration

Mechanical vibration is defined as oscillatory motions excited by varying dynamic forces. When the varying forces change in magnitude only, they are called reciprocating forces. When the varying forces change in direction only, they are called rotating forces. Vibratory motions may be further classified into two categories: steady-state and transient. The former are induced by periodically varying forces operating over comparatively long intervals of time. These periodic forces are generated by engines, machines, shafting, propellers, and other items. The vibration tests described in BuShips Specification 40T9 (SHIPS), superseded by MIL-T-17113 (SHIPS), superseded by MIL-STD-167 (SHIPS) (20 December 1954), are in the steady-state vibration category, and are analogous to fatigue-type tests of materials. The latter, transient vibrations, may be caused by shock excitation whereby impulses of energy may be applied with varying degrees of suddenness and duration. These excite the elements of a structure into vibrations at their natural frequencies which continue until damping forces consume the vibrational energies and static equilibrium is restored.

Shock

Mechanical shock may be defined as a rapid transfer of energy to a structure. This results in motions of the structure which are called shock motions. There are many kinds of shock motions, different in nature because of different types of excitation, such as explosions in air, underwater explosions, gunfire, impact, and sudden changes of velocity or direction of motion, and different because of differences of structures. Since no structural system is rigid, but has resilient or elastic properties which involve the ability to store and release energy, there will be transient vibrations superimposed on rigid-body shock motions. These transient vibrations will occur at the natural frequencies of the different parts of the structure. Mechanical shock may then also be defined as any phenomena producing transient vibrations in a structure.

Vibration Isolators

Vibration isolators are resilient mountings used to support a machine or other equipment in order either to reduce the transmission of steady-state vibration forces to a supporting structure, or to reduce the transmission of steady-state vibration forces from the structure to the equipment. Equipments mounted on vibration isolators are low-frequency systems compared with frequencies to be isolated. The mounts therefore transmit only a fractional part of the vibration forces of much higher frequency motions. The isolators allow comparatively large deflections under a static load, so the terminology "soft mounts" is well suited.

Shock Mounts

A shock mount is a resilient mounting which is intended to reduce the maximum accelerations transmitted to an equipment when the structure supporting the equipment is subjected to a shock motion. Naval shipboard shock mounts are comparatively stiff when compared to vibration isolators. Natural frequencies of equipments on shock mounts are generally above 25 cps. This results in an amplification of the steady-state vibrations for frequencies up to a value of 1.41 times the natural frequency (2).

Generally, the characteristics of shock mounts and vibration isolators are incompatible with shipboard applications. A vibration isolator under high-impact shock will "bottom," i.e., the flexible element (because of its low-energy-absorption capacity) will tranverse its clearance within the housing at a comparatively high velocity, and then collision between stops will occur. To avoid this, many vibration isolators have snubbers. The use of soft mounts usually results in a more severe shock than if no isolators had been used. (Of course, if an unlimited amount of clearance were available, and no problems of mechanical alignment existed, soft mounts could be used for shock and any degree of isolation could be obtained.) Load-deflection curves for typical shock mounts and vibration isolators having the same load ratings are shown in Fig. 1.

VIBRATION AND SHOCK SIMULATION

Vibration

Environmental conditions of shipboard vibration are simulated in the laboratory by vibration machines. There are three principal types in popular use at the present time, which may be classified as follows:

a. Mechanical Direct-Drive
b. Reaction
c. Electrodynamic

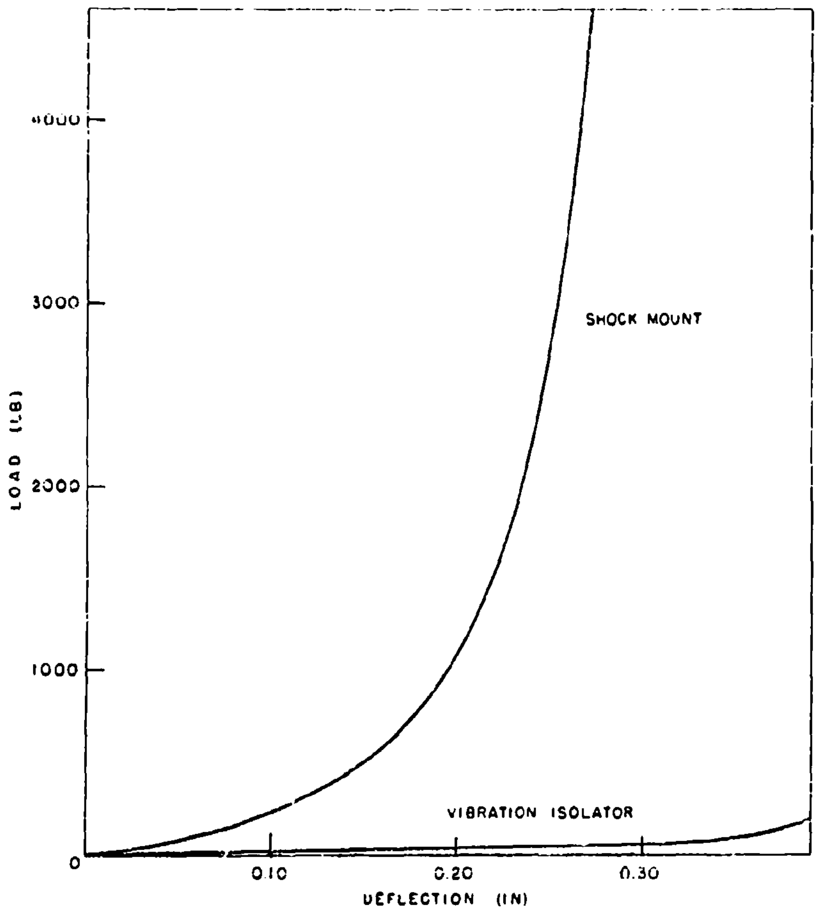

Fig. 1 - Load-deflection curves for a typical shock mount and a typical vibration isolator, both of which have a nominal load rating of approximately 20 lb. The shock mount is resonant at 25 cps and the isolator at 8 cps.

The direct-drive is sometimes referred to as the "brute-force" type, and uses a motor connected to a shaft with suitable eccentrics for amplitude adjustment. These eccentrics drive a table to which the equipment is secured. The basic displacement waveform of these machines is designed to produce sinusoidal motion, but the quality of the waveform is dependent on clearances of moving parts, bearing roughness, and on the elastic strains of the linkages and table. Records of vibration-machine table motions (3) indicate that these mechanical phenomena quite markedly affect the waveform. Figure 2 shows waveforms for a good-quality direct-drive machine which is capable of 2-inch displacement amplitude. Displacement-time curves for one cycle of motion were derived from the directly recorded records of velocity or acceleration. The maximum values of the derived displacement-amplitude time curves agreed closely with the measured displacements as determined with a traveling microscope. In most cases, the resulting displacement-time curve appeared as a simple sinusoid, as would be expected from the lack of high-frequency components in the velocity traces. The acceleration waveforms, however, had a preponderance of high-frequency components. The distortion was, of course, most apparent on the zero-amplitude runs when the machine was set to scale zero without regard for the residual motion of the table. For these records, the expanded amplitude scale further accentuates the absolute deviation.

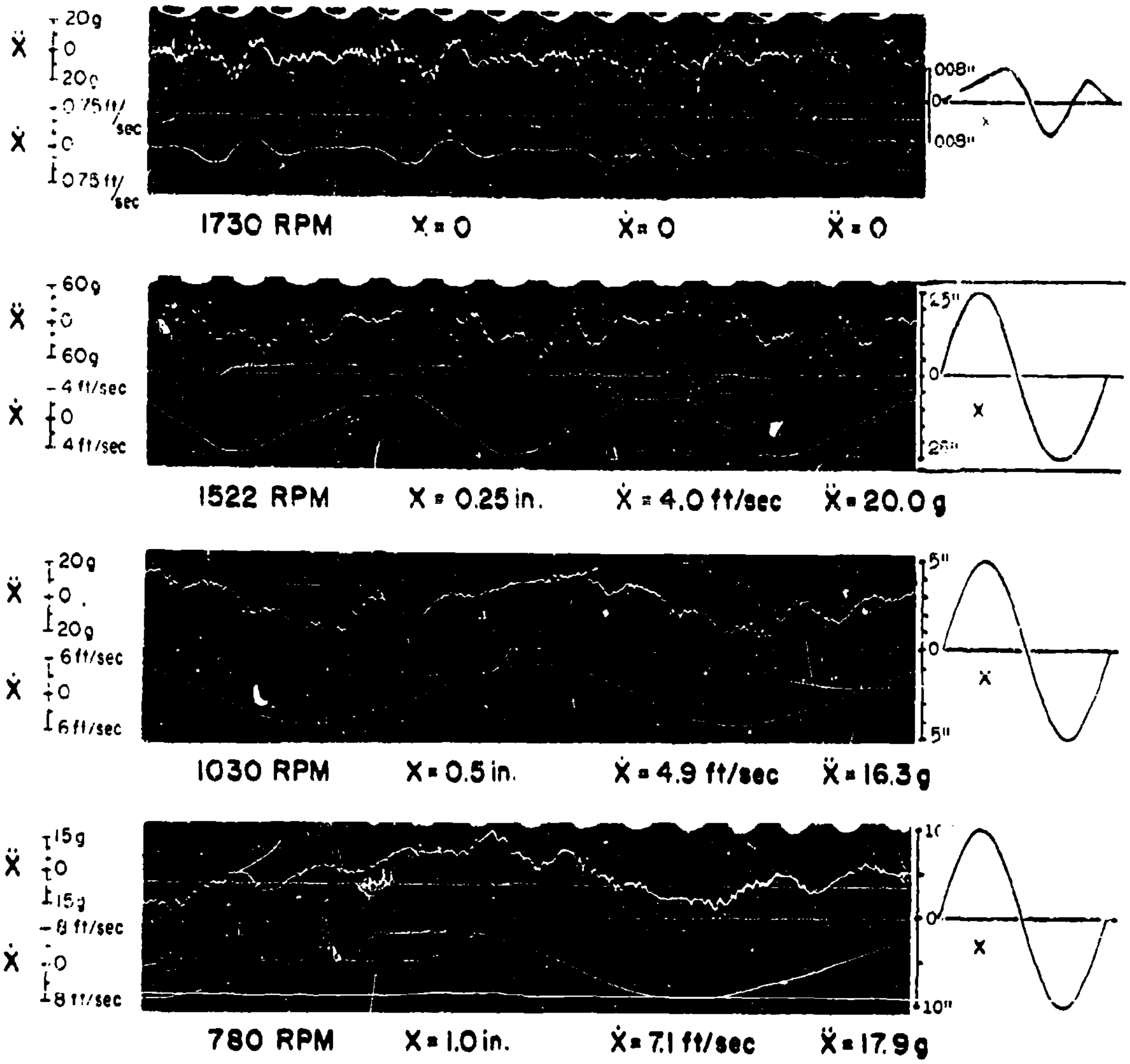

Fig. 2 - Typical test records of vertical motion of a high-amplitude vibration machine table with a 138-lb load

The larger direct-drive type machines usually require a low-frequency isolating block to isolate their vibrations from the building in which they are installed. In this case, the machine is mounted on a heavy mass, usually made of reinforced concrete, which in turn is mounted on springs so as to have natural frequencies below 5 cps for all modes of vibration.

The reaction-type machine consists of one or more motor-driven unbalanced masses, or force generators, secured to a suitable table or platform. The platform is mounted on relatively soft springs, so natural frequencies of the system are below about 5 cps. The force generators involve the use of unbalanced rotating weights to generate the vibrating force. These weights can be arranged in different ways so that the force vector can remain fixed in any desired direction, or it can be made a rotating vector. Since the reaction-type machine is mounted on a low-frequency spring suspension system, it does not require a special isolating system to isolate its vibrations from the building in which

it is installed. Unless special controls are incorporated in the design of reaction-type machines, the vibration amplitude may vary considerably when resilient-mounted equipments are being tested. This is due to the vibration-absorber effect of the equipment as it approaches and reaches resonance conditions on its mounts; the table and equipment then behave similar to two rigid masses attached together by flexible members.

The electrodynamic type of vibration generator is often referred to as the "loudspeaker type," and consists of an armature coil suitably placed in a magnetic field. The force output of the generator is proportional to the current through this coil. The input to the coil may be provided by an electronic oscillator or by motor-generator sets. Recent developments in this field have increased the force output to about 25,000 lb. This type of generator can produce frequencies as high as 10,000 cps, although larger units generally are limited to 500 or 2000 cps. They are used principally in applications of aircraft and missile testing. For shipboard environmental simulations, the mechanical types (direct-drive or reaction), with an upper frequency limit of about 60 cps, are quite adequate.

Shock

The importance of shock phenomena, as related to military operations, was greatly emphasized during the early naval battles and maneuvers of World War II. The British Navy had suffered severly from the effects of noncontact underwater explosions caused principally by German mines. Although in some cases the ships involved were not sunk, they were completely immobilized as a result of damage to propelling machinery, condenser water scoops, fire-control and communications equipment, or other parts vital to the operation of a fighting ship. A research program was initiated by the British to investigate the effects of noncontact underwater explosions on naval vessels and their equipments, as well as to determine the nature and magnitude of the forces involved. Shock machines were designed and built which simulated these conditions. Among the first of such developments was the shock machine for evaluating lightweight equipment (up to 250 lb). The U. S. Navy, working closely with the British Navy, initiated a shock and vibration program at its own activities. This program led to further developments of the lightweight shock machine, and in addition the mediumweight shock machine was developed for testing equipment to 4500 lb in weight.

A fundamental characteristic motion of the center-of-mass of the anvil of the Navy high-impact shock machines is a sudden velocity change in the order of 10 ft per sec in a time of 1 to 2 milliseconds. The accelerations of a rigid load, mounted on standard structural-channel arrangements, as required by the specifications, is in the range of 50-150 g's, and is associated with frequencies in the range of 55-70 cps for the load on the channels. More detailed studies involving actual experimental calibrations of high-impact shock machines have been described in Refs. 4 through 7. Drawings showing general arrangements of the lightweight and mediumweight machines are included in specifications MIL-T-17113 (SHIPS) and MIL-S-901B (NAVY). These specifications indicate height of hammer-drop for a given total table load of equipment, mounting channels, and bulkhead supports. MIL-T-17113 (SHIPS) is an interim military specification describing shock and vibration tests for electronic equipment, while MIL-S-901B (NAVY) is generally applicable to machinery and electrical items for shock tests. The American Standard Specification for the Design, Construction, and Operation of Class HI (High-Impact) Shock-Testing Machine for Lightweight Equipment, Z24.17-1955, can be obtained from the American Standards Association. The drawings showing details of the testing machine are also available from the American Standards Association.

DESIGN PRINCIPLES

The fundamental considerations in the structural design of electronic equipment for shipboard environments of mechanical shock and vibration are stiffness, or high natural

frequencies, and lightness. For purposes of shipboard-equipment design, frequencies above 35 cps may be considered high natural frequencies. Appendix A elaborates on this theme and, by the use of some examples involving the application of static and dynamic forces to simple beam structures, attempts to demonstrate analytically the reasons for support of the stiffness-and-lightness approach to the design of shipboard electronic equipment. In addition, some conclusions pertaining to the concepts of underdesign and overdesign, as related to weight reduction, are brought into focus with efficient design. Subsequent sections, dealing with practical and specific design considerations, are based on the results of developmental evaluations (as described in the introduction to this manual), which also confirmed this approach.

The advantages of stiff structures for electronic equipments can be briefly enumerated as follows:

a. For normal types of construction, maximum stresses induced by shipboard type shock and vibration are less for stiff than for flexible members.

b. Relative motions are reduced between parts and components, thus preventing collisions.

c. Fatiguing or breaking of wires and shafts is minimized.

d. The smaller required clearances permit a more compact design.

e. Greater exciting forces are necessary to cause damage or failure at resonant frequencies for the higher frequency systems.

f. The higher modes of vibration are more difficult to excite when a structural member is stiff.

It is especially necessary to design stiff structures if an equipment is to be shock-mounted. As indicated previously, shock mounts cause amplifications of the steady-state vibrations for frequencies up to a value of 1.41 times the natural frequency of the flexibly mounted equipment. If the main structure or the secondary structures, principally chassis or other supporting members, have natural frequencies in the same range as the shock-mounted unit, the vibration amplitudes of these components are further increased. The vibration amplitude of a structure is proportional to the product of the amplification factors of the equipment on its mounts and of the structure on the equipment. It is not recommended that vibration isolators (low-frequency mounts) be provided for items within a shock-mounted equipment, because of the excessive vibration amplification which occurs when the forcing frequency coincides with the natural frequency of the vibration-isolated item, and also because of the possibility of collision under high-impact shock. Larger internal clearances are required, which increase size and weight of the equipment. For these reasons, it is better to provide stiff secondary structures with natural frequencies high compared with those of the shock-mounted unit.

Another important and elementary consideration is that the structure to which the mounts are attached should be much stiffer than the shock mounts, or the structure may deform more than the shock mounts. It is not difficult to increase stiffness. The results of adding a little additional material in the right places of an equipment are gratifying.

Since clearance and accessibility for servicing of electronic equipment is of paramount importance and must remain unimpaired after severe shock, plastic deformation must be restricted to a minimum. Therefore, it appears logical to design the equipment structure for shock so that the maximum stresses do not exceed the elastic limit or yield point of the materials. However, experience indicates that minor structural plastic deformations observable by visual inspection can generally be tolerated. It is recommended

that the maximum design stress for shock conditions be the yield point of the structural materials. The factor of safety for this design criterion relies on minor structural plastic deformations and greater strengths under loads of short duration.

The interrelationship of steady-state vibration and shock phenomena is important in the design of shipboard electronic equipment, and both must be considered together. This is emphasized in the following discussion. In ships, there are two principal causes of vibration. One is propeller-blade excitation, and the other is the unbalanced forces of propeller and shafting. The vibration excitation of the hull structure caused by the propeller consists of water-pressure variations against the shell plating as the propeller blades rotate. The frequency of these pulses is equal to the number of blades of the propeller times the rotational speed of the shaft. For a conventional four-bladed propeller rotating at a maximum speed of 375 rpm the forcing frequency would be 375 × 4 or 1500 cpm (25 cps). Some recent designs of submarine propellers are five-bladed, and exciting or forcing frequencies of 2000 cpm (33 cps) are expected. As previously mentioned, vibration isolators, with excursion limits restricted as they are for shipboard installations due to space considerations, are generally excluded from shipboard electronic-equipment application because of their bottoming action under high-impact shock. Their application would be more detrimental to the equipment under shock conditions than solid mounting. To restrict the travel of presently installed equipment under shock forces, relatively stiff mounts are used which amplify or aggravate vibrations in the frequency range (5 to 33 cps) common on ships. Resilient mounts of this type are called shock mounts. In order to minimize the vibration-amplification effect, the lowest natural frequencies of parts of an equipment and of the equipment on its mounts (resilient or solid) should be sufficiently high to insure that it will occur above the forcing frequency range (some exceptions may occur if noise reduction, or other overpowering reasons, require low-frequency mounting systems). If the equipment will not be used in small craft or submarines, the upper frequency range will be reduced to about 25 cps.

The situation has required, particularly for base-mounted equipment, the removal of the resilient shock mounts and substitution of solid spacers in their place. This was necessary because, even though the stiffest available resilient mounts were used, the natural frequencies of the rocking modes of vibration were still below the maximum test frequency. The excessive rocking motions caused malfunctioning and failures, both mechanical and electrical, of the equipments. The solid mounting raised these natural frequencies above the test range so that at the maximum frequency of the test range the motion amplification was small. This reduced the vibrational motions of the equipments to a point where satisfactory operation could be obtained. The equipments could be made sufficiently rugged for survival under shock test. This situation has frequently occurred, and it is emphasized to indicate that the mounts in these circumstances become so hard that they are less flexible than the structure. The next step is to achieve shock protection by relying on the "structural resilience" of the equipment.

In the case of mediumweight electronic equipment, especially in the weight range between 500-2500 lb, the rocking modes are even more troublesome when resilient mounts are used. The section on cabinet and console design elaborates on this problem. However, a mounting design which has been successfully applied to this problem is described here. The arrangement consists of common types of base mounts selected to provide adequate stiffness in the vertical direction. The transmissibility ratio or vibration amplification in this direction should be kept to a low value of about 2.5 to 1. In conjunction with this resilient base mounting, two or more flex-plate bulkhead stabilizing brackets are employed. Figure 3 shows details of two designs of such a bracket, which are flexible in the vertical direction and stiff in the horizontal direction. This horizontal stiffness is very effective in raising the natural frequencies of the rocking modes excited by horizontal vibration. These flex-plate brackets can be easily fabricated from 1/8- or 3/16-in.-thick 1020 cold-rolled flat stock. A recommended minimum bend radius is twice the thickness of the plate so that bending stresses will not become excessive. They should not be used

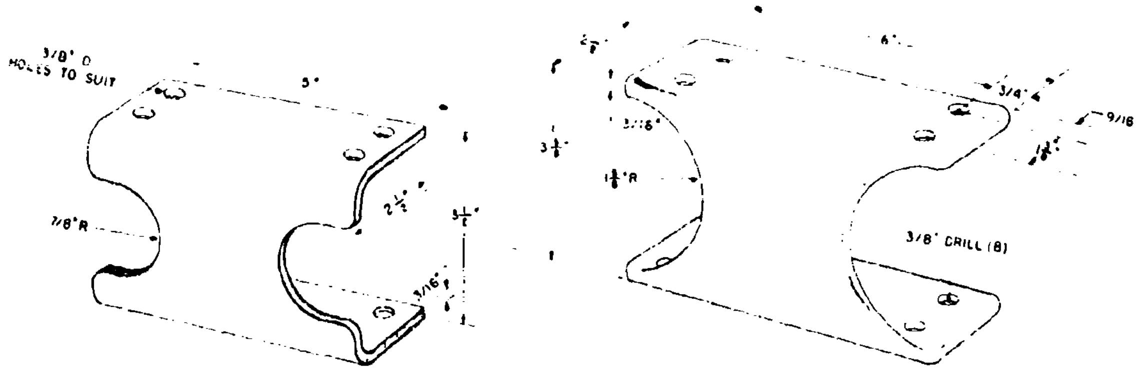

Fig. 3 - Two designs of bulkhead flex-plate mounts
(NRL Flexmount, Forkois)

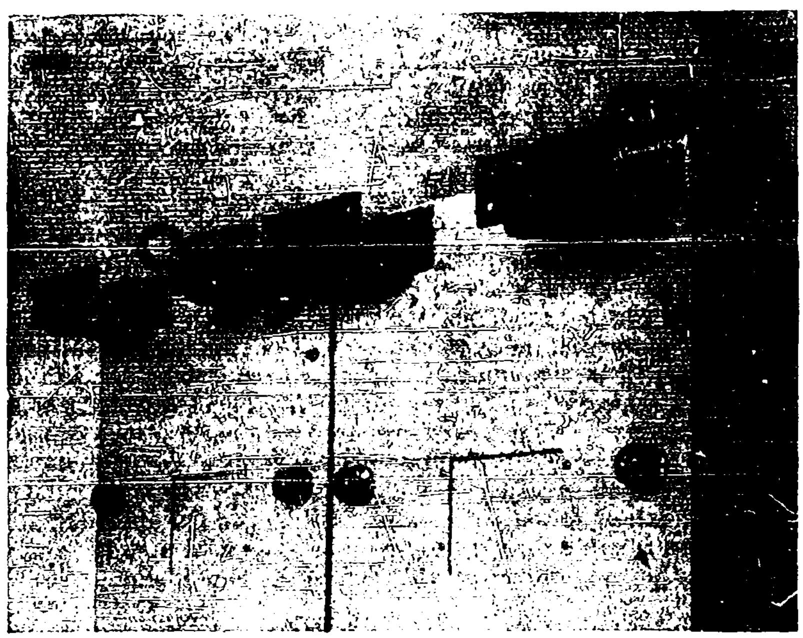

Fig. 4 - 2000-lb dual radio transmitter with 4 flex-plates
and rear view of 4 adjustable dowel pins

unless the vertical transmissibility factor is low (less than 2.5) because of probable fatigue failure. Figure 4 shows a successful application of four flex-plates to a 2000-lb radio transmitter.

A summary of the main design factors and conclusions advanced in this section is given as follows:

a. The basic structural design considerations for electronic equipments for shipboard environments of mechanical shock and vibration are stiffness and lightness. Natural frequencies of all structural members should be above 35 cps.

b. Greater shock and vibration reliability is attainable by the use of stiff structures with natural frequencies above 35 cps, than with flexible structures, i.e., structures with natural frequencies below 35 cps. This statement also applies to the structural design of the electronic components or parts supported by the equipment.

c. Vibration isolators should not be used for well-designed electronic equipments intended for naval-shipboard installation. The use of vibration isolators (low-frequency mounts) is more detrimental to electronic equipment under high-impact shock than solid mounts. Exceptions may be made for nonlinear mounts of special design involving abnormal clearance.

d. The great majority of electronic equipments in the lightweight category (up to 250 lb) can be properly and economically designed without excessive weight and without the need of shock mounts. The term "solid mounting" as used here means the utilization of rigid nonresilient spacers to support the equipment at the locations where shock mounts are usually applied. This method relies on the "elastic structural resilience" of the equipment for shock protection.

e. With regard to mediumweight electronic equipment (above 250 lb and up to about 3500 lb), the application of solid mounting has been considered and used with caution because of equipment complexity. The heaviest piece of equipment with solid mounting which was considered satisfactory for shock and vibration by this laboratory was a 700-lb radar-repeater switchboard. The use of resilient base mounts in conjunction with steel flex-plate bulkhead brackets appears at this juncture to be a satisfactory arrangement for both shock and vibration.

The assembly techniques and structural shapes and sections which provide stiffness, strength, and lightness use box or tubular sections for beams and columns, ribbed panels and edge-flanging for flat plates, edge-flanging for lightening-holes, gussets and brackets for end connections of beams and columns, and stiffeners for large load-supporting plates or surfaces. This is detailed in the sections on chassis, and on cabinets and consoles.

MATERIALS

In selecting structural metals for environments of shock and vibration, the most desirable properties are high ductility and high yield point. In cast materials, an effort should be made to obtain suitable castings with elongations of standard specimens of not less than about 5 percent. The three important structural metals in general use are steel, aluminum, and magnesium. For shipboard environment, steel and aluminum are preferred. Specification MIL-E-16400 Ships for electronic equipment indicates the particular steel and aluminum specifications applicable to construction parts. Magnesium requires specific approval of the bureau or agency concerned for each application. The modulus of elasticity (Young's modulus) is a very important property, since it is a measure of the inherent stiffness of materials. Young's modulus for steel is nominally 30×10^6 psi, for aluminum 10×10^6 psi, and for magnesium 6×10^6 psi. Although aluminum is one third the density of steel, the actual saving in weight (when designing for equal structural stiffness, or equal structural natural frequencies) is not in direct proportion to the density, because of this difference in Young's modulus. If equal stiffness is used as a design criterion, steel structures less than 50 percent heavier than aluminum structures are possible.

Stainless-steel bolts, in spite of their high tensile strengths, frequently have very low yield values. Unless stainless-steel alloys are selected and heat-treated to provide good elastic properties, it is better, from a shock-strength viewpoint, to use high-tensile-strength bolts.

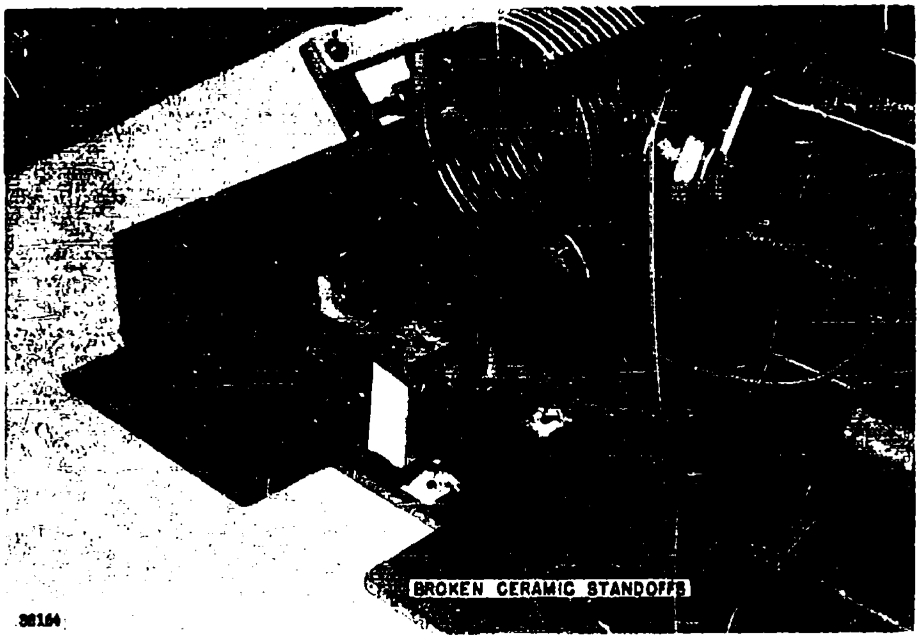

Fig. 5 - Broken ceramic standoffs caused by shock test

The use of ceramics for insulating material should be avoided wherever possible, because of their brittleness. Figure 5 shows two broken ceramic standoffs which supported a vacuum variable capacitor in a high-voltage rf circuit. These were replaced with standoffs made of silicone fiberglass laminate which passed the test successfully.

CABINET AND CONSOLE DESIGN

Introduction

Previous literature deals in general with the mathematical approach to design and excludes information and design criteria of an empirical nature. This section presents, without mathematical elaboration, those design practices which under tests have proven to be successful for shock and vibration. Once a proper concept of what constitutes good design is envisioned by the individual, the necessary formulae can be obtained from other sources.

Geometry and Mounting of Cabinets

One of the early decisions which must be made in a proposed design concerns the physical geometry of a cabinet required to house the various components of a system. The dimensions of equipment for shipboard service are usually closely defined by restrictions created by hatches, passageways, or access openings, through which the units must pass, and by space allocations. Within these limits, however, certain dimensional ratios have been established as an upper boundary beyond which vibration difficulties multiply rapidly. Since some of the lightweight and most of the mediumweight equipments are mounted on resilient mounts, it is important that the geometrical shape of the final cabinet design favor the struggle to obtain sufficiently high equipment-mount frequencies. This is particularly important for rocking modes of vibration.

Vibrations of the Equipment-Mount System—The lowest structural or mount resonance normally desired is 25 cps (MIL-T-17113), since the exciting frequencies existing on the average vessel are 23 cps or less. A later specification (MIL-STD-167 (SHIPS), 20 December 1954) requires vibration table amplitudes of 0.030 in. between 5 and 15 cps, 0.020 in. between 16 and 25 cps, and 0.010 in. between 26 and 33 cps. It allows exceptions for equipment intended for installation on a particular class of vessel if it is known that the frequency range on this class will not include the upper specified range. This specification indirectly requires either that the lowest equipment-mount resonance be several cycles per second above the test frequency range, or that means be provided to keep the transmissibility factor to tolerable values (less than about 3) within the test range.

Mount locations are commonly at the base, or at the base and upper rear sections of an equipment. For some directions of exciting vibration, the equipment will rock on the mounts so that extreme amplitudes occur at locations most distant from the supports. In general, if the mounts are located unsymmetrically with respect to the center of gravity of the equipment, then all modes of vibration of the equipment-mount system will be excited by a simple linear forcing vibration (8, 9). If however, the mounts are located symmetrically with respect to the center of gravity, or in some cases if mounts are used with their axes suitably inclined, it is possible to make the different modes of vibration independent of each other. They are said to be uncoupled. Under these conditions, linear exciting vibrations will not excite the rocking modes. This is a very desirable condition. While it is not practical to attain this condition in most cases, it is often possible to approach it much more closely than is usually done.

Common Mounting Arrangements—The two most frequently used shipboard mounting arrangements for electronic units are those in which mounts are either attached only to the base or bulkhead panels of an equipment, or attached to the unit's base with sway mounts attached to the upper rear panel of the cabinet. The first arrangement is used almost exclusively on lightweight equipments, whereas mediumweight units employ the second method.

Figure 6 is presented to show poor equipment shape, from a mounting viewpoint, and the corrective measures required to compensate for it. The original intention of the designer was to have the sway mounts attached only to the upper rear panel of the cabinet. This produced a system which, because of the extreme depth of the unit, had severe unbalance in the horizontal side-to-side direction. For vibration applied in this direction, a low rocking resonance occurred well below the maximum sustained test frequency of 23 cps even with the stiffest commercially available mounts installed. Since it was not possible to obtain resonances above this frequency using the intended mounting, or to obtain tolerable vibration levels, the arms shown in the figure had to be employed. This approximately decoupled all modes of vibration. Sufficiently high resonant values were obtained at the expense of much wasted space. The same results could have been achieved by locating the sway mounts on the top panel of the cabinet in the same vertical plane, but a hangar extending either down from the overhead or out from the bulkhead would have then been necessary. Unless the use of these braces is foreseen prior to the acceptance tests, it is sometimes possible that the structure of a cabinet in these areas may not be able to withstand, without damage, the inertial loading created by the braces. A different shape of the equipment could have been the solution to the problem.

Figure 7 shows a frequency meter whose height was much greater than the minimum distance between the centers of the base mounts. When horizontal vibration was directed parallel to the generator's front panel (the shorter horizontal dimension), a rocking mount resonance resulted low in the spectrum of test frequencies. Because of the high degree of ruggedness possessed by the equipment, it was possible to substitute metal feet for the shock mounts, and by so doing raise resonance above 23 cps. The cabinet of the unit was considered too flexible to effectively use sway mounts without extensive modifications.

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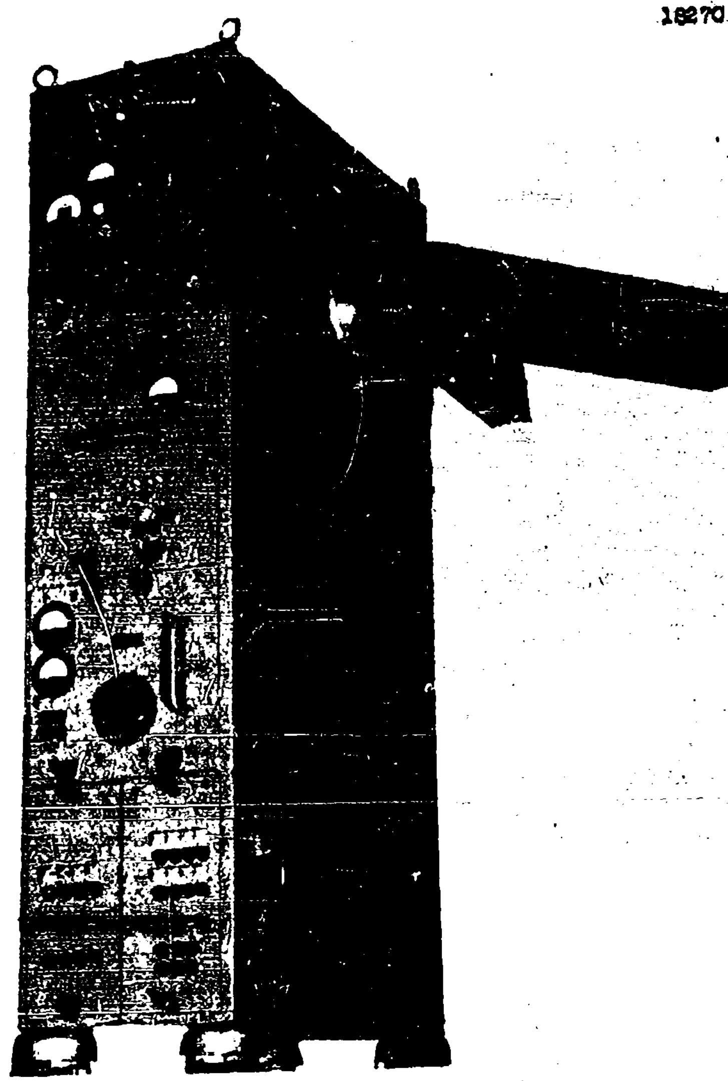

Fig. 6 - Poor equipment geometry from a mounting viewpoint

Empirical Design Factors—What constitutes a ratio of equipment height to width above which solid mounts are required for base mounted equipment? Equation (1), taken from Ref. 10, expresses the lowest horizontal rocking frequency of a base shock-mounted equipment (four mounts) in terms of the vertical translational mount frequency and a factor Q, determined by the physical characteristics of the unit.

$$\omega = Q\sqrt{\frac{4K}{M}}$$

$$Q = \sqrt{\frac{1}{2}\left(1 + \frac{A^2}{R^2} + \frac{B^2}{4R^2} - \sqrt{\left[\frac{1}{2}\left(1 + \frac{A^2}{R^2} + \frac{B^2}{4R^2}\right)\right]^2 - \frac{B^2}{4R^2}}\right)} \qquad (1)$$

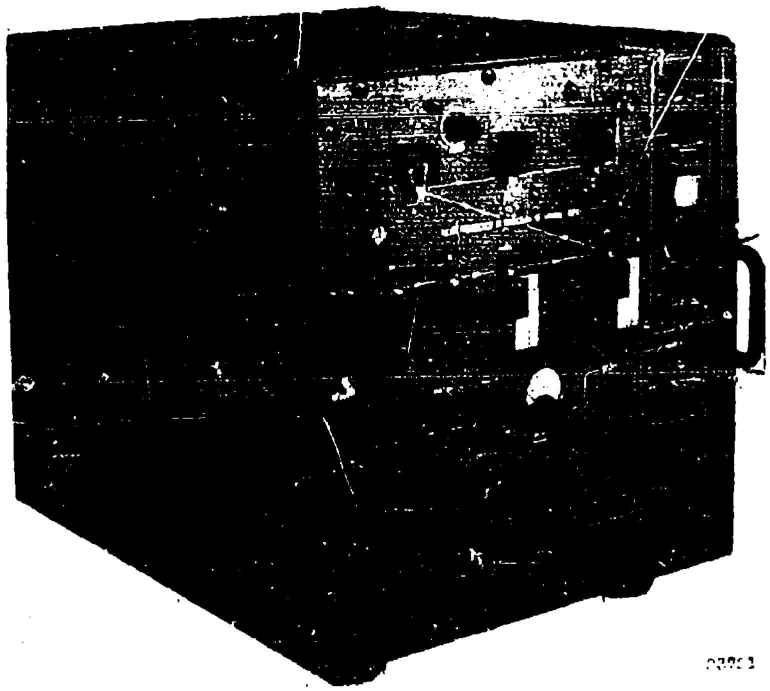

Fig. 7 - Frequency meter whose height is much greater than minimum distance between centers of the base mounts

where

ω = lowest horizontal rocking frequency

A = height of the center of gravity above the base

B = minimum distance between the centers of the shock mounts

K = mount stiffness ($K_v = K_h = K$)

M = mass of equipment

R = radius of gyration about an axis through the center of gravity, perpendicular to the direction of vibration.

The factor Q considers the radius of gyration about the longer horizontal axis passing through the unit's center of gravity, the minimum distance between the centers of the shock mounts, and the height of the center of gravity above the base. The assumption is made that the mounts have an equal stiffness in both the horizontal and vertical directions.

For a group of eight typical base-mounted units submitted to the Navy for acceptance tests, Q from Eq. (1) versus a ratio of the equipment's height to the minimum distance between the centers of the base mounts (2A/B) was plotted in Fig. 8. The type of mounting (solid or resilient) was determined by the actual vibration acceptance tests, i.e., solid mounts were used if resilient mounts were unsatisfactory. All of the units were originally submitted with resilient mounts by the manufacturers. In applying the equation

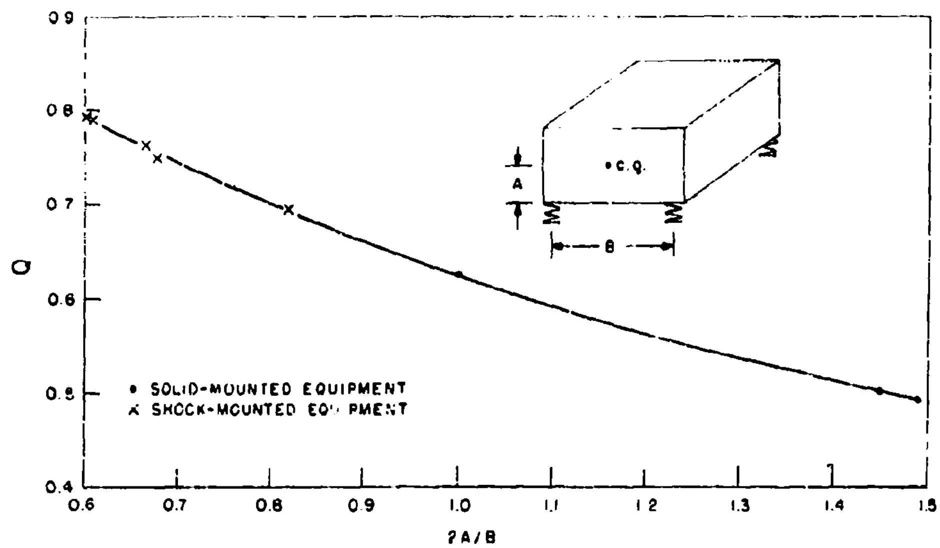

Fig. 8 - Graph showing relationship of Q (Eq. 1) versus ratio of equipment height to minimum distance between centers of base mounts

to these real equipments, the radius of gyration, R, was determined on the basis of an even distribution of mass throughout the volume of the cabinet, since actual values of R had not been determined experimentally. A persistent effort was made to obtain satisfactory vibration characteristics with shock mounts on all of the units concerned, but even the stiffest commercial mounts available did not raise the lowest rocking frequency of some units above the spectrum of test frequencies. As the graph shows, units having a 2A/B ratio of unity or more required a solid mounting. Obviously form alone did not determine the resulting rocking frequencies, but form was an important determinant.

For base-mounted units which must be shock mounted, the value of unity is normally accepted as an upper design value for 2A/B. It should be evident that higher rocking frequencies are more easily obtained as this ratio decreases. This value can be exceeded for solid-mounted or sway-mounted equipment.

The difference between actual and calculated values of the horizontal rocking frequencies of the plotted equipments are given in Table 1. These values are based on the original mounting system determined by the manufacturer. The differences, which range from 3 to 14 cps, are caused primarily by unknown cabinet flexibilities, together with some error due to assumptions made relative to mass distribution.

TABLE 1
Rocking Resonances of Base-Mounted Equipment Plotted in Graph 1

No.	Q	2A/B	Calc. Freq. (cps)	Actual Freq. (cps)	Diff. (cps)
1	0.49	1.49	20.4	15.0	5.4
2	0.50	1.45	15.4	12.0	3.4
3	0.63	1.00	15.1	11.0	4.1
4	0.69	0.82	39.0	29.0	10.0
5	0.74	0.68	46.8	33.0	13.8
6	0.77	0.67	29.2	26.0	3.2
7	0.79	0.60	33.0	29.0	4.0
8	0.79	0.60	29.8	25.5	4.3

The stiffness of the mount to be installed on the equipment should be taken as approximately 1.4 times that of the stiffness determined from Eq. (1). This factor, which compensates for cabinet flexibility, is an average value based on the units considered in Table 1 (and the assumptions made relative to them), and applies to typical electronic equipments of good general construction with relatively even mass distributions. It does not necessarily apply to mediumweight units or units having different mount arrangements. The two units in Table 1 having widely divergent calculated and actual frequencies were not considered in determining this factor.

For resiliently mounted units requiring stabilizing or bulkhead mounts, a ratio of unity for cabinet depth to the distance between the centers of the stabilizing mounts is considered from experience to be an upper value for design for normal mass distribution. Greater values usually result in rocking resonances below 25 cps. Figure 9 shows a radar transmitter-receiver (72-in. tall) which had approximately this value, and for horizontal vibration parallel to the front panel, resonance occurred at 24 cps, even though the stiffest available cup-type mounts were used. If this depth ratio is kept at or below 0.75, the problem of obtaining rocking frequencies at or above 25 cps is much easier.

The natural frequency calculations of flexibly mounted systems are presented in Refs. 8 and 9. Because of the laborious mathematics involved in the solution of systems (particularly one plane of symmetry), nomographs have been used.

Mass Distribution

Suitable form, by itself, will not insure high rocking frequencies, since these frequencies are also a function of the radius of gyration and the location of the center of gravity. Therefore, the designer must concern himself with the proper positioning of components to bring the center of gravity as close as possible to the planes of the mounts.

Basically, the problem is resolved by mounting the heavy components as close to the planes of the mounts as possible. Since power transformers normally constitute the heavier elements in most electronic units, the lower areas of the cabinet should be reserved for them and other massive components. This is illustrated in Fig. 10. The upper section of the cabinet was bolted directly to the transformer housing. This design not only produced a low center of gravity, but it greatly simplified the problem of obtaining adequate structural stiffness.

Fig. 9 - Radar receiver-transmitter (72 in. high) with a ratio of unity for cabinet depth to distance between centers of the bulkhead stabilizing mounts

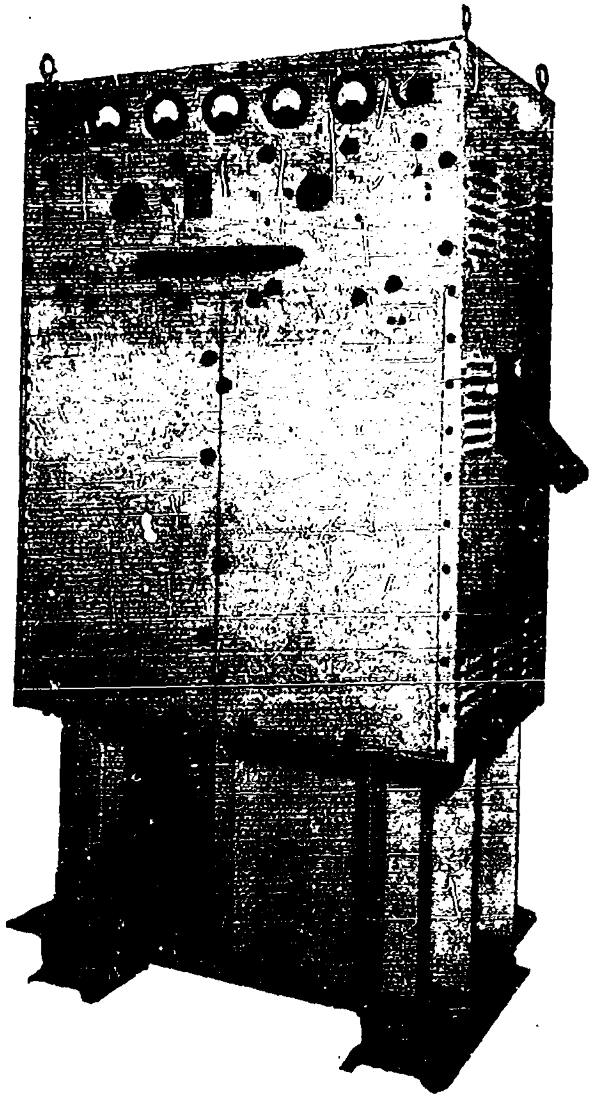

Fig. 10a - Radar modulator cabinet, showing upper section bolted to transformer housing, doors closed

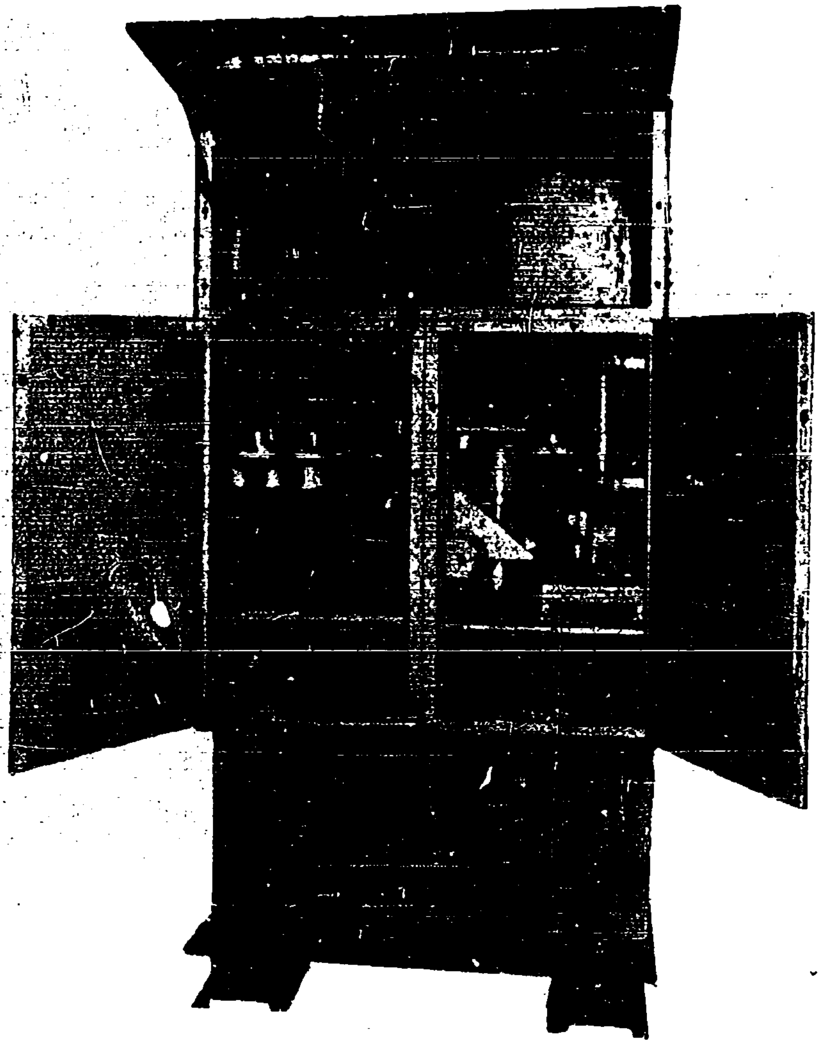

Fig. 10b - Radar modulator cabinet, showing upper section bolted to transformer housing, doors open

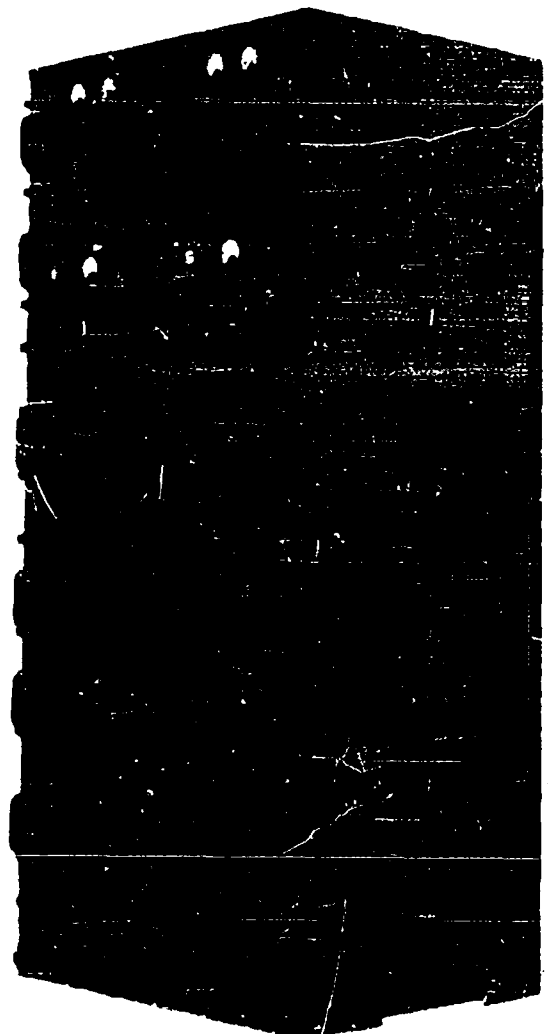

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Fig. 11 - Radio transmitter showing conventional drawer arrangement, with power supplies located in bottom three drawers

The dual transmitter in Fig. 11 better illustrates properly located power supplies of a more conventional nature. In the three lower chassis levels, immediately above the air intake screens, the modulator and the low-, medium-, and high-voltage power supplies were concentrated, thus eliminating the need for power transformers in the upper chassis. Power transmission circuits supplied the requirements of individual components. Even with this good arrangement, the lowest rocking frequency of this exceptionally dense equipment (total weight, 2060 lb) occurred at 24 cps in the horizontal direction parallel to the front, only one cycle per second above the maximum sustained test frequency of 23 cps.

Figure 12 shows an interior view of the Frequency Meter of Fig. 7, a lightweight equipment, and illustrates good mass distribution in that the heavy components are mounted low in the unit.

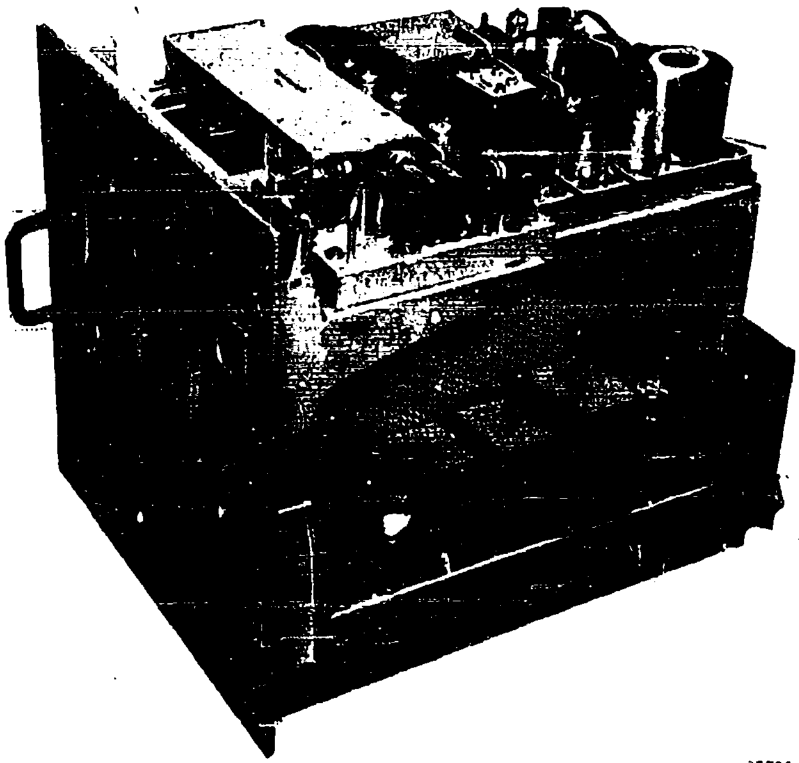


Fig. 12 - Interior view of frequency meter shown in Fig. 7, illustrating good mass distribution

Other more unconventional systems are sometimes found in naval equipments, but, in any event, proper mass distribution and good dimensional ratios are long steps toward the production of a design that meets shipboard vibration and shock requirements.

Structural Design

After establishing the approximate shape and weight distribution of a proposed equipment, the problem of a properly designed structure presents itself. Although each individual unit usually exhibits distinctive qualities peculiar only to itself, certain other physical characteristics must be shared by all if damage-resistant equipment is to result.

Requirements of Structures—The structures of shipboard equipment must possess a stiffness for torsional, or rocking, as well as translational, modes of vibration, because of the "whipping" motion experienced by sway-mounted equipments. Low structural resonances seldom cause difficulty from vibrations directed perpendicular to planes which contain the mounts, for base, bulkhead, or combination base and bulkhead mounting systems. It is relatively easy to obtain high frequencies in these directions. However, for vibrations directed parallel to planes which contain the mounts, rocking modes are excited which are most serious, and any lowering of the resonant frequency by structural flexibility may be intolerable. Evaluation of actual equipment submitted for Navy acceptance tests shows that this problem is prevalent in many current equipments in the mediumweight class.

The vibration specifications require vibration to be directed along each of three mutually perpendicular axes. This makes the problem of torsional rigidity of the structure of considerable importance. The effective use of gussets at joining points of structural members is usually a solution. Figure 13 shows a mediumweight equipment (rear view) whose structure lacked torsional stiffness to properly resist horizontal vibration directed parallel to the front panel. The mounting arrangement for this system consisted of four base and two sway mounts (not shown). Although a large number of gussets were used for stiffness, they were missing in the most effective locations, specifically in horizontal planes. The lowest rocking resonant condition was raised from 21 cps to 23 cps by virtue of the added gussets indicated by arrows and by aluminum plates installed between the four lower chassis by riveting to the horizontal frame members. Since resonance occurred at 23 cps, the maximum sustained test frequency, the amplified vibrational forces caused severe damage and occasioned the rejection of the equipment.

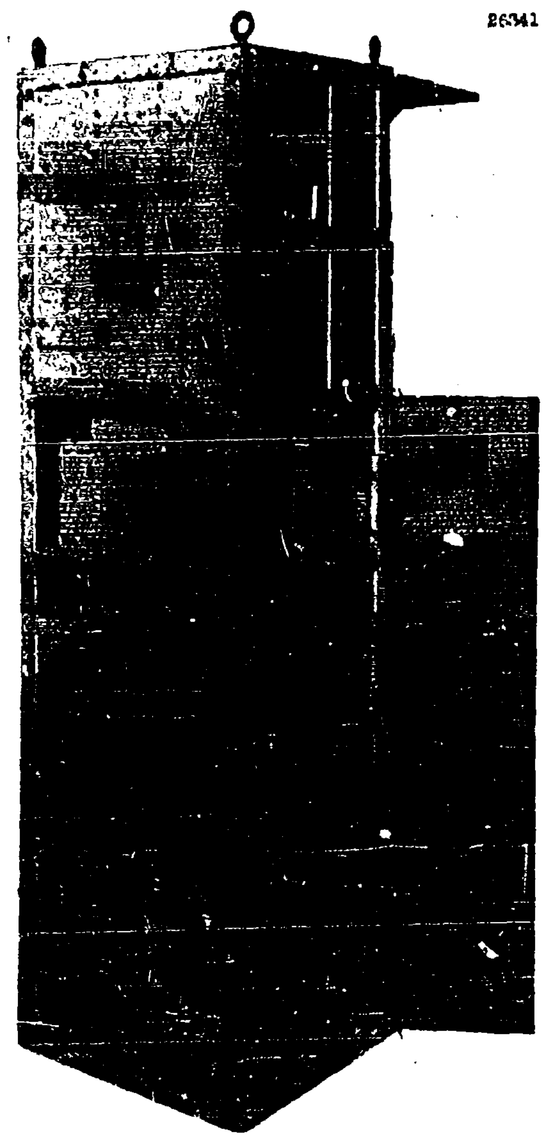


Fig. 13 - Multichannel transmitter whose structure lacked torsional stiffness

Light and Mediumweight Construction—The line drawings in the following section show some effective, practical ways of obtaining stiffness in a design. Because of different structural requirements, medium- and lightweight equipments will be taken up individually, although there are, of course, many points of similarity between the two classes. The dividing line apparently falls in the vicinity of 150 to 200 lb, and is determined by needs of the cabinet. Units weighing more than 200 lb generally require a frame to support the various elements of the system, particularly where more than one chassis is involved. Loads created by the chassis or components are transferred to the frame; from the frame they are passed on to the mounting system. The cabinets of equipments having only one or two chassis and weighing less than 200 lb do not necessarily require a frame, provided the loads imposed by the chassis are transferred as directly as possible to the mounting system. In the latter case, the skin of the cabinet itself can be formed in a manner that will produce adequate stiffness.

In mediumweight units, the structure is further stiffened when the metal skin or covering is fastened over the frame. Laboratory observations of the same loaded frame, partially covered and uncovered, have shown a resonant frequency difference of 2 or 3 cps; this increased stiffness is produced by the development of shearing stresses in the skin. The monocoque construction techniques used in aircraft fuselages provide an excellent example of this shearing stress effect. Sheets of thin aluminum are wrapped around formers placed at intervals along the length of the fuselage, and by loading the formers, this type of construction is capable of carrying enormous loads.

The term "stressed-skin" has been applied in particular to equipments whose cabinets were constructed without a frame. Several "stressed-skin" mediumweight cabinets have been tested under laboratory conditions and, after being modified structurally and mounted so as to not greatly excite the rocking modes, have been found satisfactory for shipboard requirements. However, they do not generally possess the stiffness and ruggedness desired for shipboard equipment. These cabinets are usually constructed by forming, out of the skin itself, a series of box sections to which the loads are applied. The thickness of the material out of which these box sections are formed seldom exceeds 0.10 in. The additional weight added to the larger equipments, by the use of a properly designed frame, appear to provide benefits that compensate for the weight increase.

Mediumweight Structures—The most common structural shapes used in frame construction are the simple angle and channel. Other shapes having "T" or "Z" cross sections are used but they are confined in general to a particular equipment, or relegated to a special application within an equipment. Because of good strength and stiffness properties, steel is a preferred material, but objections can be raised against its use from a weight and corrosion standpoint. With the exception of the very heavy units, aluminum frames have been and are being used successfully in electronic equipments weighing as much as 1500 lb. Since resonant frequency or stiffness problems are not quite so critical in most lightweight equipments, aluminum is used almost exclusively in them. Substituting aluminum for steel does not, however, effect a weight reduction of two thirds. For similar members of equal stiffness, the section modulus of the member using the lighter material has to be increased, with a consequent reduction in the weight saved. Although some weight reduction is realized, its magnitude depends on the loads involved.

Joints and Gussets—Figures 14, 15, and 16 illustrate simple, efficient, and effective construction joints. In Fig. 14, the angles forming the frame are shown butted together prior to the installation of gussets. More elaborate butt joints could be used, but they are not necessary strengthwise because of the addition of gussets. Depending on fabrication procedures used by the manufacturer, gussets can either be riveted or bolted (Fig. 15), or welded (Fig. 16) to the frame. Any of these joining methods (or combinations) are effective, but the advantages or disadvantages may differ in production. To permit unobstructed installation of the skin or covering over the frame, protruding welds must

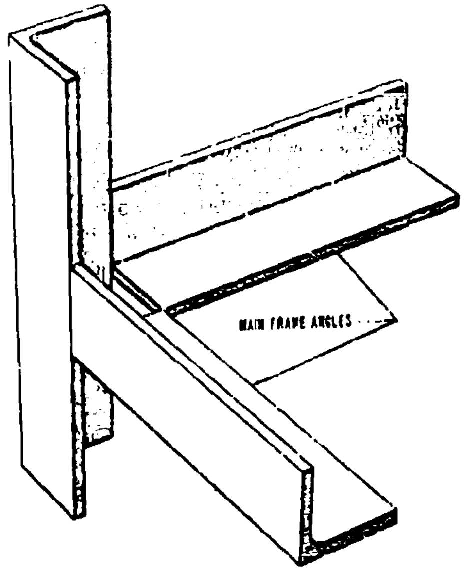


Fig. 14 - Angle frame arrangement prior to installation of gussets

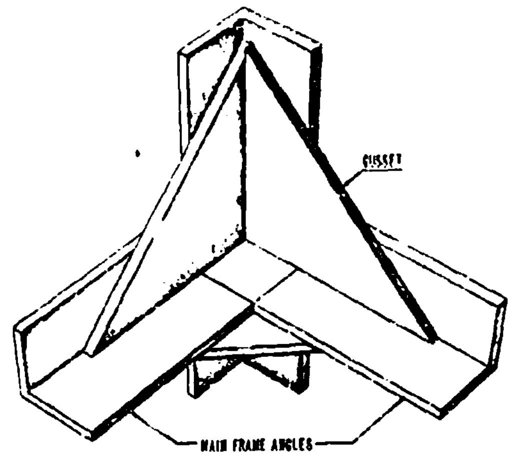


Fig. 15 - Gusset arrangement for bolted or riveted construction

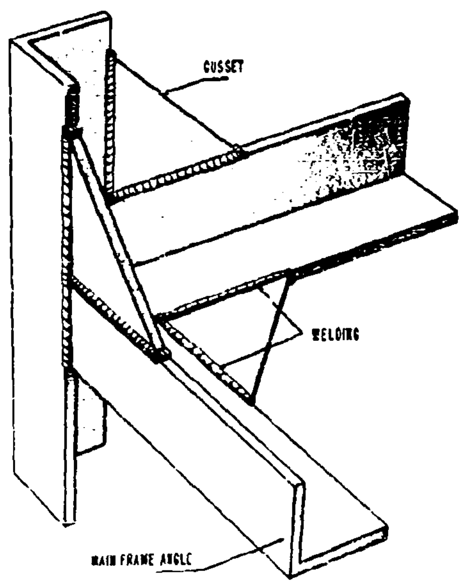


Fig. 16 - Gusset arrangement for welded construction

be ground flush with the outside surfaces. The thickness of the gussets should be approximately equal to the thickness of the frame members. By gusseting in the three mutually perpendicular planes, desirable stiffness characteristics are incorporated into each joint with a very small increase in overall weight.

Corner joints are pictured in Figs. 17 and 18. Those gussets located at the four top corners of the frame can be "heavied up" to provide a load bearing surface to which eye-bolts, for handling purposes, can be attached.

When members of other shapes are used, the same illustrated techniques are still applicable.

Joining Practices—Rivets and/or bolts are the most common fasteners employed in joining the various members together, and they consistently do a good job in proper designs. The welded structures demonstrate excellent ruggedness and, in many respects, are superior to riveted or bolted designs. In riveted or bolted structures, the main members could be tack-welded prior to gusset installation, to minimize jigging problems.

Spotwelds can be used successfully to fasten the metal skin or covering to the framework, especially since recent improvements in spotwelding techniques have increased their strength and fatigue properties. But because of the magnitude of the forces involved

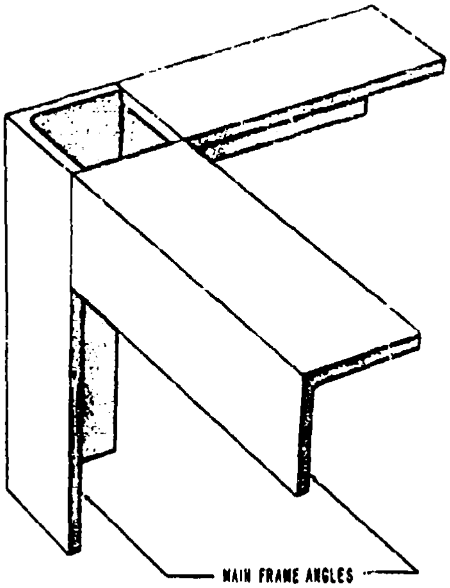


Fig. 17 - Angle corner-joint arrangement without gussets installed

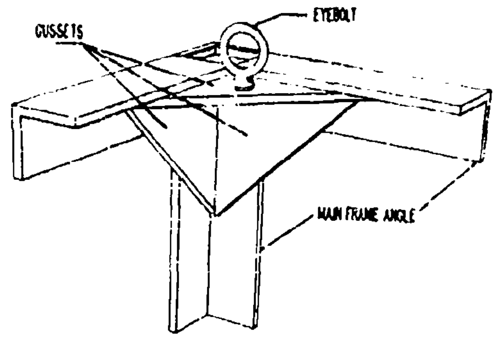


Fig. 18 - Angle corner joint with gussets installed

for shipboard use, the frames of the heavier mediumweight equipments should not rely on spotwelding. Spotweld damage is aggravated by the fact that a high stress concentration exists in the junction between the two bonded materials; under repeated loadings, this increases the possibility of damage.

Torsion Plates—Aside from the stiffness offered by corner gussets, it is also possible to introduce torsional stiffness into a cabinet by the installation of a thin metal plate fastened to the frame between the chassis. Shear and tension stresses developed in the plate resist any change in the geometry of the horizontal cross section of the structure. To prevent excessive vertical vibrations of the plate, a shallow channel can be welded diagonally across the plate as shown in Fig. 19. A reduction in stiffness may result when diagonal surface discontinuities (coined stiffeners) are used in thin plates, even though stiffness perpendicular to the plane of the plate is realized, because under load, the depressions or discontinuities tend to deform (Fig. 20). Plates 1/8 in. thick or less will probably satisfy the requirements of most equipments.

For systems in which air circulation is needed for cooling, cross braces could be substituted for the plates. Various other methods of bracing peculiar to specific equipments could be used instead, but some form of bracing is necessary for high rocking resonances in the larger equipments. For the smaller units (under 500 lb), the gussets usually provide enough stiffness to make extra bracing unnecessary.

Shock Mount Installation—Since many mediumweight and a few lightweight equipments require shock mounts, an understanding of the problems involved in their installation is important. For vibration, stiffness is again an important factor. It is desirable to have the resonant frequency of the unit's structure at least twice that of the mount. In the larger mediumweight units, this desire is not often realized because of practical engineering limitations, but the smaller equipments present no real problems. Unfortunately, the effectiveness of a generally stiff structure can be reduced sharply by excessive flexibility of the surfaces to which the shock mounts are attached. This is another of the major causes contributing to low rocking resonances.

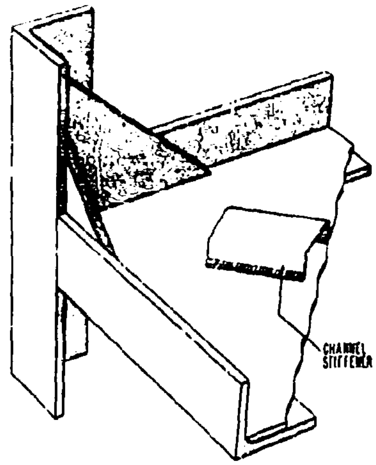


Fig. 19 - Torsion plate or divider with welded channel stiffener

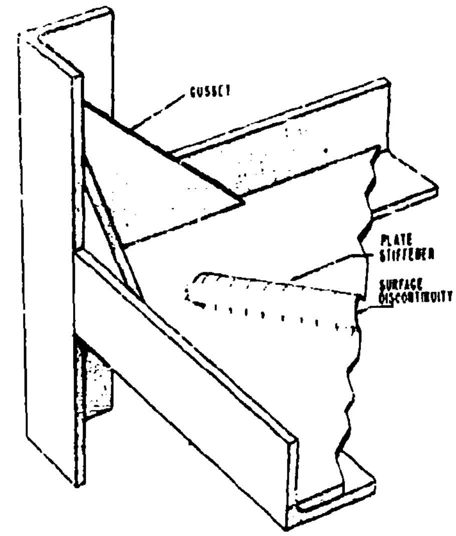


Fig. 20 - Torsion plate or divider with coined stiffener

Since the loads imposed on the structures by the mounts are, in a sense, concentrated, they must be distributed over the largest possible area of the frame to minimize stresses. One very effective construction for the installation of base shock mounts on mediumweight equipment is shown in Figs. 21 and 22, and consists of channels and suitably gusseted angles attached to a base plate. The mounts are installed in the channels.

The greatest portion of the inertial loads, created by shock, must be transmitted to or from the base through the four vertical frame members. Therefore, high shearing forces can exist in the gusset fasteners if too few fasteners are provided. For this reason, welding is very desirable in the fabrication of the base assembly, since greater resistance to shear results. Buckling of the structure frequently occurs at the base of equipments if the inertial forces are not properly distributed. To achieve proper distribution, channels should be incorporated into the design. On the lighter units, the channel extending between the outboard mount channels (Fig. 22) can often be eliminated, but good practice, regardless of weight, calls for channels under the mounts. Depending on the materials and loads involved, base plates approximately 1/4 in. thick, when properly reinforced, appear to be adequate for the average mediumweight equipment.

For bulkhead mounts, an arrangement similar to that shown in Fig. 23 can be employed. By using a channel section under the mount, support is provided by two horizontal and one vertical member, with the result that good load distribution is effected. Lightweight units with sway mounts do not require such elaborate bracing in the superstructure. Obviously, individual situations might require that some of these suggestions be modified, but the importance of sufficient stiffness of the surfaces to which the shock mounts are attached cannot be overemphasized.

Mounting of Chassis—Those directions of vibration causing the most damage to the average equipment are the horizontal (side-to-side) and vertical directions. For sway-mounted equipments, horizontal (front-to-back) vibration is less damaging. Chief among the causes of damage for vertical vibration are low chassis resonances often aggravated

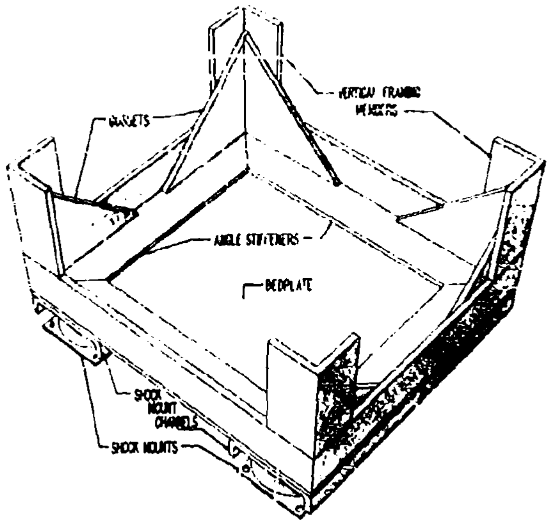


Fig. 21 - Stiff construction for the installation of base shock mounts on mediumweight equipment, top view

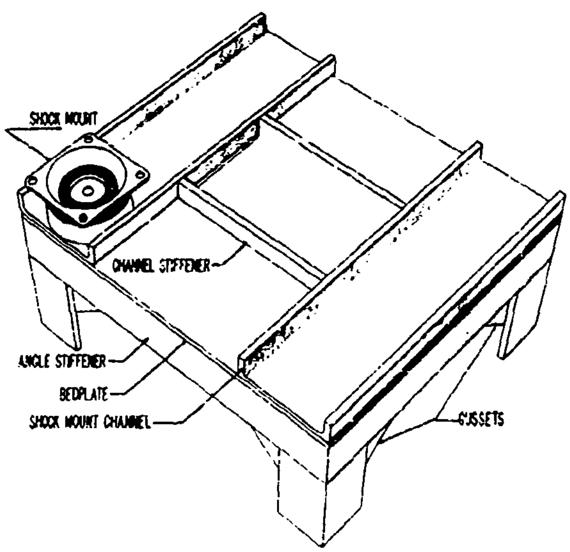


Fig. 22 - Bottom view of construction shown in Fig. 21

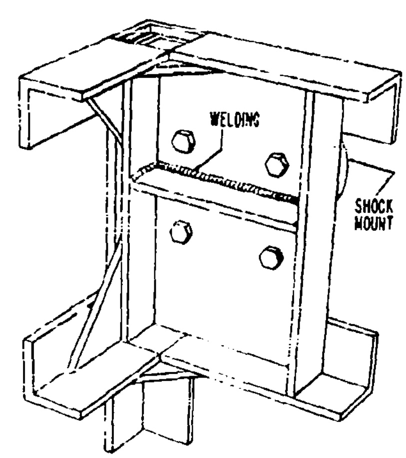


Fig. 23 - Structural reinforcement for bulkhead shock mounts for mediumweight equipment

by the absence of the outlined design features. Although design practices relative to chassis themselves are treated in another section, mounting requirements must necessarily be discussed here, since they directly involve the cabinet structure.

Two major requirements must be met in a properly mounted chassis. Under vibration and shock, the chassis must be restrained from bouncing, and the inertial loads created by forces acting on the chassis should be transferred directly to the frame of the cabinet.

Figures 24, 25, and 26 show the normal way of satisfying both requirements. The rear of the chassis is supported by guide pins, fastened to the cabinet's frame; these pins slide into corresponding receptacles attached to the chassis. The front of the chassis is bolted to the frame by reduced-shank screw-type fasteners, located around the periphery of the front panel. To ease alignment difficulties, the front panel fasteners lock into "floating" nuts attached to the frame.

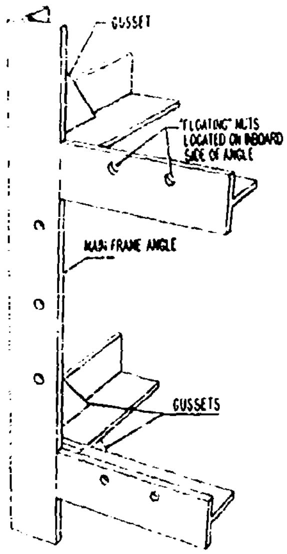


Fig. 24 - Frame construction showing clearance holes for drawer or chassis front-panel thumbscrews

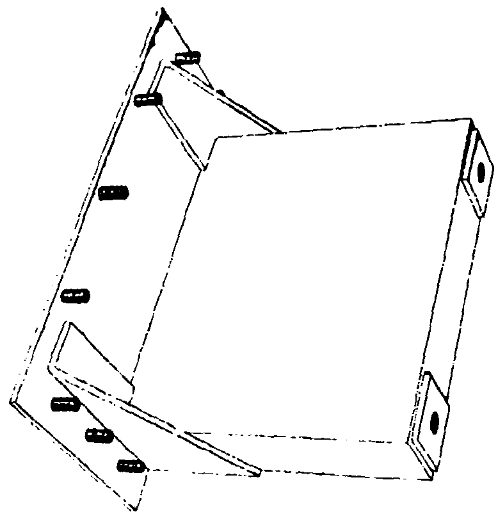


Fig. 25 - Good chassis construction, showing front panel thumbscrews and hole-plate reinforcement for dowel pins in the rear

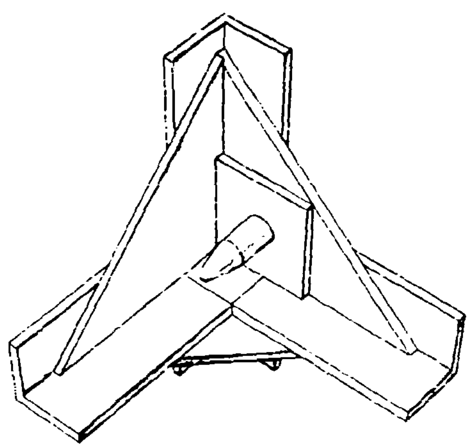


Fig. 26 - Dowel pin arrangement for riveted or bolted gusset

Because of severe abrasion, the tapered guide pins must be made of steel. They usually vary from a diameter of 1/4 in. to 1/2 in. Stubby pins (no longer than 1 in.) minimize bending stresses, consequently little space should exist between the receptacle and the shoulders of the guide pins. The receptacle should also be steel, with a free sliding fit occurring between the pin and the receptacle. It is usually better practice to locate the pin on the frame of the cabinet and the receptacle on the chassis, since greater stiffness results. Figure 26 illustrates a pin attached to a riveted or bolted gusset, and Fig. 27 shows one installed on a welded gusset; where chassis slides are used, the welded arrangement is more economical of space.

The problem of providing close-fitting dowels (maximum desired tolerance on diametral fit, +0.005 in.) is a serious production problem, particularly where interchangeability of chassis drawers is a requirement. Figure 28 shows a method for providing an adjustable chassis dowel pin. Two 1/2-in. diameter pins were used successfully on chassis weighing about 120 lb, and provided a maximum adjustment up to 3/16 in. Figure 4 shows this installation for the chassis of a 2000-lb radio transmitter.

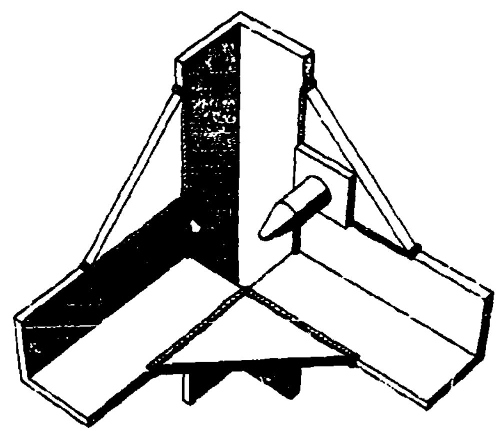


Fig. 27 - Chassis dowel pin arrangement for welded gusset

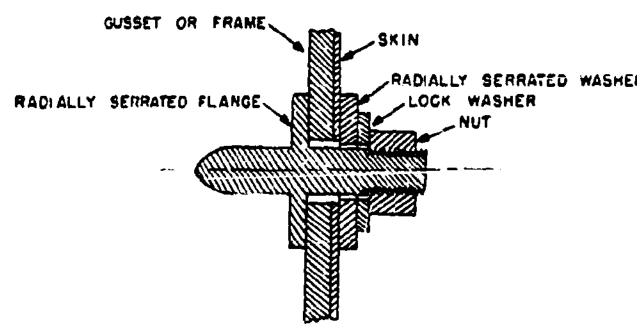
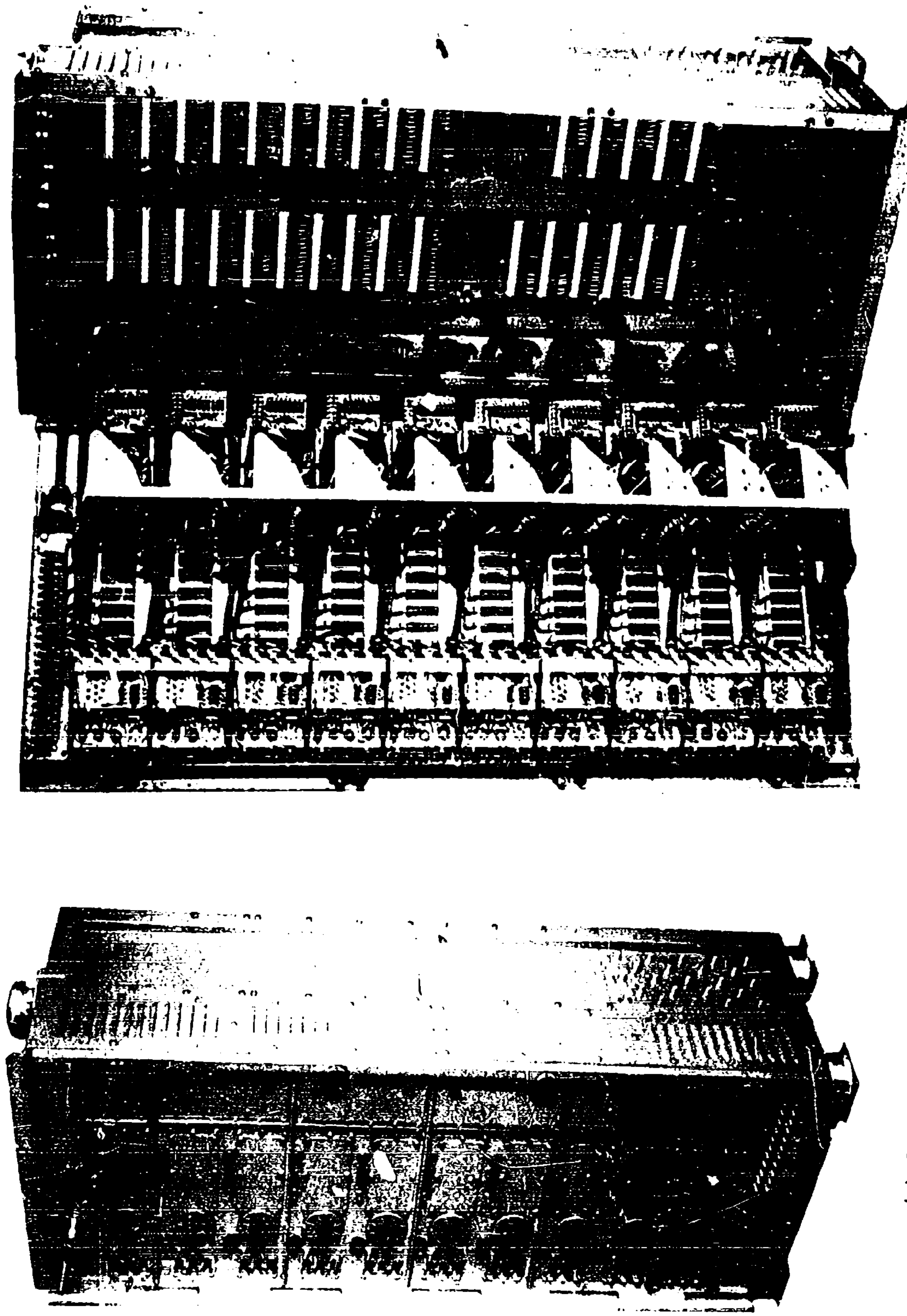


Fig. 28 - Adjustable chassis dowel pin

It is important to place fasteners at regular intervals around the entire periphery of the braced front panel, otherwise a low resonant condition for front-to-back vibration can result. For convenience in repair or trouble-shooting operatings, chassis slides of the tilting or nontilting variety can be used, but in the "locked-up" position, they must not be required to carry large inertial loadings.

"Swing-Out" Construction--For improved accessibility, some equipments are designed with a large percentage of the parts mounted on a door or panels. The radar switchboard in Fig. 29 is typical of such equipments. Two possible detrimental features of this type of design relative to shock and vibration should be observed. First, the design moves the center of gravity frontward in the unit, making it difficult to obtain high rocking frequencies. In the case of the switchboard, it was necessary to remove the mounts from the rear panel and to reinstall solid metal mounts at the center of the top panel (the mounts shown are rubber but were later changed to solid metal types). To repeat earlier statements, this sort of mounting may result in an uneconomical use of space and inconvenience in installation, although the mounting system itself is highly desirable for vibration reduction.



(a) Door closed

(b) Door open

Fig. 29 - Radar repeater switchboard with door-mounted parts

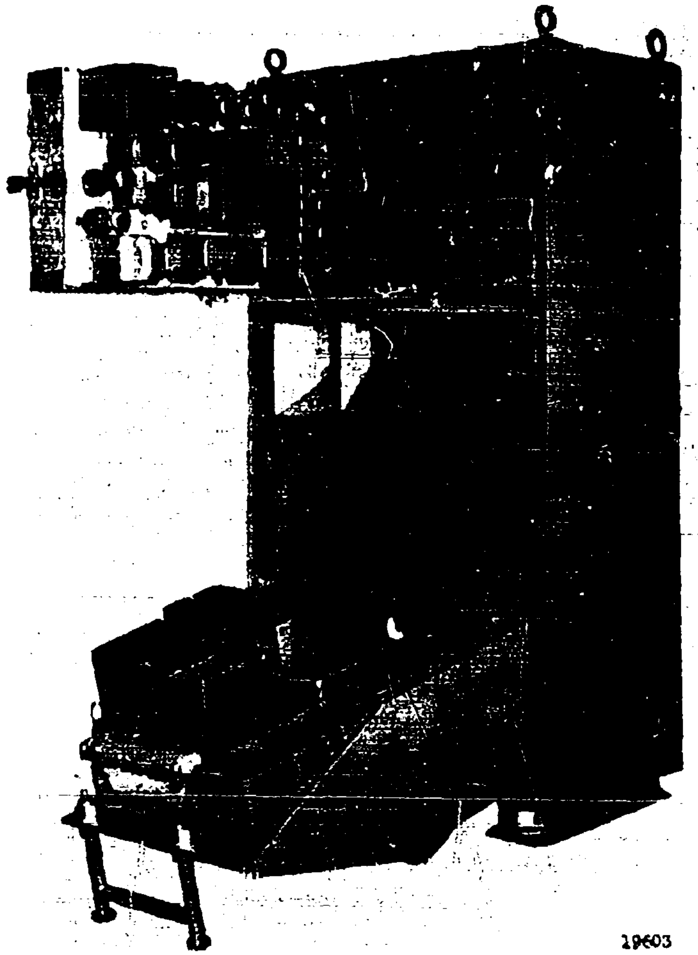


Fig. 30 - Sonar receiver-transmitter showing another arrangement of "swing-out" construction

Second, a difficulty exists in adequately stiffening the loaded door or panels to raise structural resonances above 25 cps. For this reason, the horizontal stiffeners, welded across the face of the door (Fig. 29a), were found to be necessary for suppression of vibration.

The sonar transmitter-receiver in Fig. 30 shows another arrangement of "swing-out" construction. Originally, support was provided for the lower chassis (power supply) only by the lower pivots and the two upper fasteners on which the chassis is shown resting. In order to introduce enough stiffness into the structure for vibration conditions, it was necessary to fasten two channels to each side of the cabinet. The top of the hinged power supply was bolted to the lower of these channels. These channels stiffened the frame considerably. The shear pins shown on the right side of the switchboard in Fig. 29b, which provided relief for the loads carried by the door fasteners, could have been used with good advantage on the sonar unit.

There are other objections and advantages to this type of design, but where "swing-out" construction must be used, the principles discussed earlier relative to center of gravity location and stiffness must be closely observed. In the heavier mediumweight

equipments, a design in which the components are chassis mounted or mounted on a stationary part of the cabinet's frame presents fewer problems for shock and vibration than a design where components are door mounted.

Lighweight Structures—In the lightweight equipments, cabinets can be constructed to resist shipboard shock and vibration without using frames of the type discussed for medium-weight equipments. By properly forming the skin and reinforcing load-bearing areas, enough stiffness can be designed into the cabinet to meet current specification requirements.

Figure 31 shows an example of a well-designed stressed-skin cabinet of an equipment with two chassis and weighing 183 lb. Both chassis were mounted on slides to facilitate maintenance and repair. Each slide was mounted on a box section attached to panels of the cabinet. Because of the equipment's height, sway mounts were necessary; where they were attached to the structure, triangular reinforcements were provided. Screw-type fasteners were installed around the entire periphery of each chassis front panel, which, in conjunction with the front panels of the chassis, increased the stiffness of the structure for horizontal, side-to-side, vibration. The rear of each chassis was supported by dowel or guide pins. Across the front of the cabinet a reinforced channel was installed to support the adjacent edges of the chassis.

No frame, as such, was used in the construction of the cabinet. As can be observed on the rear panel, the skin forming the side panels was flanged, and the rear panel was spotwelded to these flanges. Horizontal torsional stiffness of the structure was increased when the chassis were locked into position.

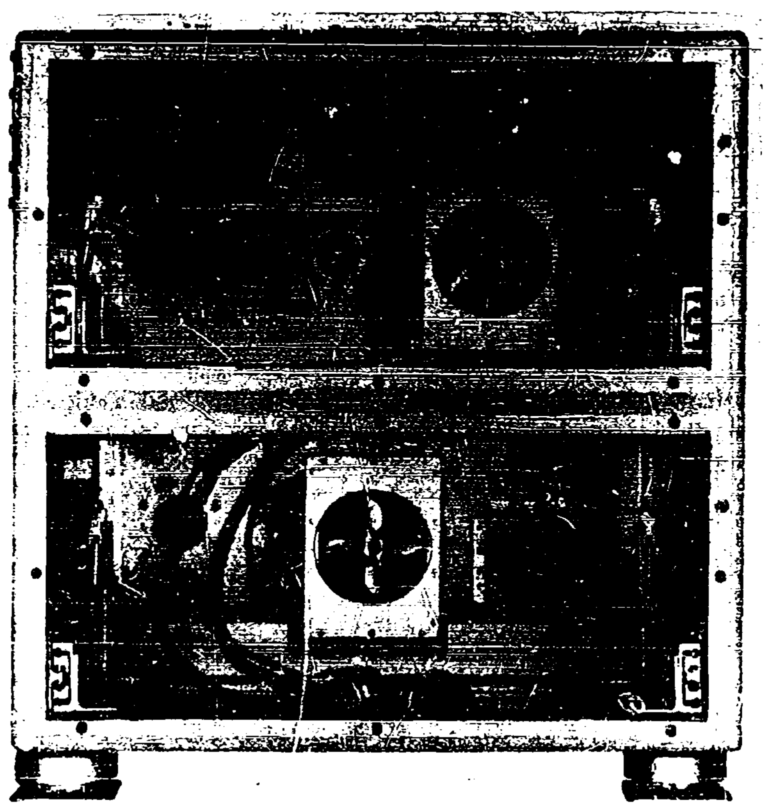


Fig. 31 - Example of a well-designed stress-skin cabinet, complete weight 183 lb

One of the problems that frequently appears in no-frame construction which is worthy of special attention is excessive flexibility of the surfaces supporting the shock mounts. The skin alone offers little stiffness. A modified version of the technique illustrated in Fig. 17, using lightweight channels, is one very effective solution. The shock mounts should be located on or in the stiffener in order to benefit.

For equipments having more than two chassis, or weighing more than 200 lb, a frame usually provides an added factor of safety, although there have been satisfactory exceptions. Lightweight equipments, in general, offer no difficulty as far as shock and vibration is concerned, provided good design practices are applied.

Structural Design Data

Mathematically exact solutions of cabinet design problems are too laborious to be practical. Solutions obtained are approximations based on the response of systems having one, or a few, degrees of freedom. To let the designer know if the approximations are in the realm of reality, Table 2 has been prepared. In the table, a number of the main structural characteristics of certain equipments which successfully passed shipboard shock and vibration acceptance tests (Military Specification MIL-T-17113) are enumerated. The size, material, and, in the case of the vertical elements, the number of the members forming the cabinet structure of equipments whose weights varied from 55 to 2060 lb, are tabulated, along with the method used to join the members of the structure together. Also presented are skin thicknesses and the method employed in attaching the skin to the frame. The number and locations of shock mounts are included to provide comparisons for future designs. The mounts were generally sufficiently stiff to cause the resonant modes of vibration to be slightly above the endurance test frequencies.

Because an equipment appears in the table, it should not be inferred that the equipment was ideal. The dimensional ratios of several were certainly not desirable; however, the structures demonstrated adequate ruggedness, and for this reason they were included. The chart does not give information regarding gussets or other supplemental stiffeners, since each unit differs in this respect; earlier suggestions must be discriminately applied.

Too often, the idea of excessive mass is associated with designs to be subjected to shipboard shock and vibration. While such designs must obviously be heavier than equipments which are not required to withstand high inertial forces, massive designs are not demanded to satisfy shipboard requirements. If the concept of designing for stiffness and lightness is accepted as a criterion, rugged equipments of reasonable weight can result.

CHASSIS

Introduction

Two aspects of the chassis design problem will be discussed in the subsequent paragraphs. First, those design practices which increase the chassis ability to withstand damage and which contribute to an improvement in the equipment's operational performance will be considered on the basis of systems previously tested. Second, at the conclusion of this discussion, a method for determining the approximate natural frequency of loaded chassis will be presented.

Evaluation tests conducted on a large number of naval electronic equipments have demonstrated the importance of good chassis design in the attainment of reliable operational performance under conditions of shipboard shock and vibration, even though permanent mechanical damage to the chassis does not occur. For the frequencies

TABLE 2
Structural Properties of Electronic Equipment which Satisfactorily Passed Shipboard Shock and Vibration Acceptance Tests

No.	Weight of Equipment (lb)	Overall Dimensions (in.)			Number of Shock Mts			Material and Size of Cabinet Main Frame Members	Method of Joining Frame Members	Material and Thickness (in.) of the Skin	Method of Skin Attachment
		Height	Width	Depth	Base	Bulkhead	Overhead				
1	55	6-7/16	17-1/2	19-1/8	4	-	-	Aluminum stressed-skin construction (single chassis)	-	Alum., 0.083	Alum. Welded
2	135	18	18-7/8	17-1/2	4	2	-	Aluminum stressed-skin construction with reinforced corners	-	Alum., 0.093	Riveted
3	146	13-7/8	19	18-1/2	-	-	-	Aluminum stressed-skin construction (single chassis)	-	Alum., 0.093	Alum. Welded
4	199	22	19-1/8	25-5/8	-	-	-	Aluminum stressed-skin construction (single chassis)	-	Alum., 0.093	Alum. Welded
5	405	42-1/2	17	22	4	4	-	Aluminum Angles Vert: 4 - 1-1/4 in. x 1-1/4 in. x 3/16 in. Hor: 1-1/4 in. x 1-1/4 in. x 3/16 in.	Riveted	Alum., 0.083	Bolted
6	550	35-1/4	37-1/8	28-1/8	4	2	-	Aluminum stressed-skin construction. The side and top panels were rolled to form 1-1/2 in. x 1 in. channels. Various supplemental stiffeners were used for reinforcement.	Spot and Heli-arc Welding	Sides, Top, and Back - Alum., 0.125 Front (doors) - Alum., 0.188	Spot-, Heli-arc Welded and Bolted
7	700	71-3/4	28-3/8	28	4	2	-	Aluminum stressed-skin construction. The side and top panels were rolled to form 1-1/2 in. x 1 in. channels. Various supplemental stiffeners were used for reinforcement.	Spot and Heli-arc Welding	Sides, Top, and Back - Alum., 0.125 Front (doors) - Alum., 0.188	Spot-, Heli-arc Welded and Bolted
8	715	43-3/4	25-1/4	30	4	-	3	Steel stressed-skin construction. The sides were formed to produce ten vertical box sections, each 3 in. x 1/2 in. Numerous horizontal supplemental angle and channel sections added for stiffness.	Riveted and Spotwelded	Steel, 0.062	Riveted and Spotwelded
9	1515	77-7/16	32	28	6	6	-	Aluminum Angles Vert: 4 - 2 in. x 2 in. x 3/16 in. Hor: Front - 2 in. x 1 in. x 3/16 in. Others - 2 in. x 2 in. x 3/16 in.	Welded	Alum., 0.094	Bolted
10	1565	68-1/4	41	28	6	2 Flex Plates	-	Cabinet divided into an upper and lower section. The structure of each section was formed from steel angles. Lower section. Vert: Front - 4 - 1-1/2 in. x 1-1/2 in. Rear - 2 - 1-1/2 in. x 1-1/2 in. 2 - 2 in. x 2 in. Hor: 1-1/2 in. x 2 in. Upper section. Vert: Front - 3 - 1-1/2 in. x 1-1/2 in. 3 - 2 in. x 2 in. Hor: 1-1/2 in. x 2 in.	Welded	Steel, 0.062	Bolted and Welded
11	2060	72	32	24	8	4	-	Steel [illegible] Vert: Rear, 4 angles - 2-1/2 in. x 2-1/2 in. Front, 4 channels - 2 in. x 1 in. Hor: Angles 2-1/2 in. x 1-1/2 in.	Welded	Sides - Alum., 0.036 Top - Steel, 0.056 Rear - Steel, 0.125	Bolted Bolted Welded

encountered on the average naval vessel, only the first vertical chassis mode of vibration is likely to be excited, and if such a resonance occurs in or near the spectrum of shipborn exciting frequencies, maloperation of vacuum tubes, relays, or other sensitive parts is probable.

The various manufacturers use a variety of design techniques, many of which are good. The following suggestions naturally should not be considered as the only approach to good design.

Typical Damages

Figure 32, showing the extended and tilted drawer of a stabilization data set (gyro mechanism), exhibits the results of a test on the Navy's mediumweight, high-impact, shock machine. Close examination of the tilted drawer will show that there are actually two individual chassis. The upper, or forward, section supported the largest percentage of the vacuum tubes, and was separately installed on shock mounts. Because of greater weight and the more rugged equipment involved, the aft chassis, supporting the transformers, was fastened directly to the drawer. Unfortunately, with the structural geometry involved, additional support could not easily be provided along the adjacent edges of the chassis. As a result of this and the light construction buckling occurred. The practice of shock-mounting electronic sections of an equipment and solid-mounting the power supplies is permissible, but care must be exercised in the mechanical design to insure adequate stiffness and clearances.

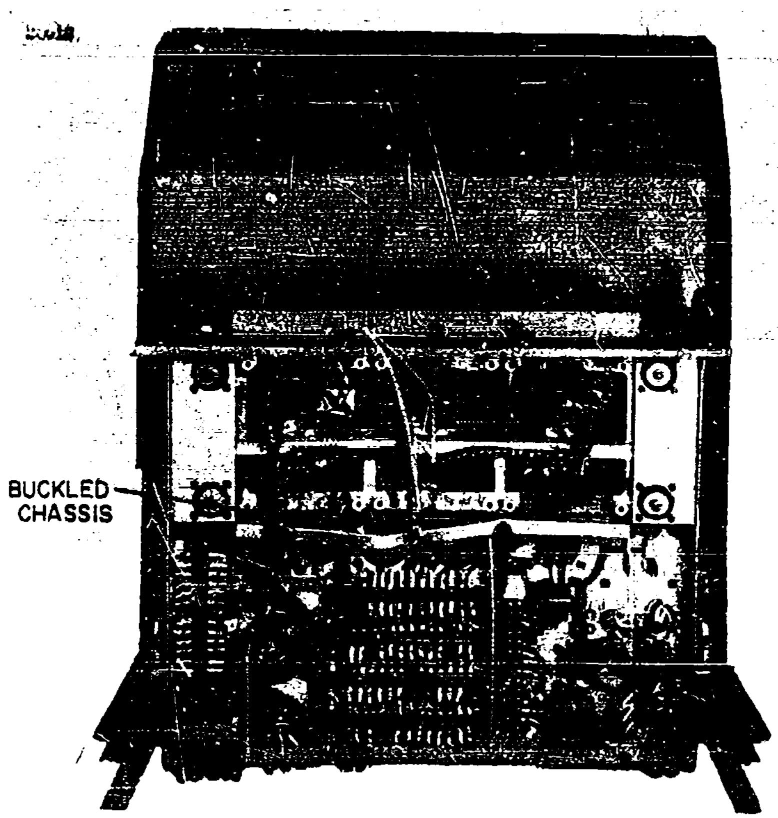


Fig. 32 - Extended and tilted drawer showing buckled chassis

Fractures initiated at points of stress concentration are a very common type of damage. Extensive laboratory tests reveal that stress concentration is the chief cause of damage resulting from vibration as well as shock. Excessive flexibility, combined with stress concentration, produces an especially serious condition for vibration. Typical chassis fractures, resulting from fatigue during vibration, occur in the bends or between cut-outs for components

Mounting Practices

Under the section "Cabinets and Consoles," it was pointed out that chassis must be prevented from bouncing during inertial loadings. This is accomplished by the use of guide pins and screw-type front-panel fasteners. Pertinent information relative to guide pin size, location, etc., is given in the earlier section. The various types of quick-release fasteners presently available are not suitable for the front-panel fasteners, although they may be used successfully on small viewing panels or in the installation of some lightweight components.

Chassis accessibility for maintenance and trouble-shooting is improved by the provision of ball-bearing slides with built-in tilting features such as shown in Fig. 33. Several types of slides permit tilting either up or down and locking at a number of predetermined angular positions. Since the slides are not designed to carry heavy loadings, the weight of the chassis in the "locked-up" position should be carried entirely by the guide pins and front-panel fasteners.

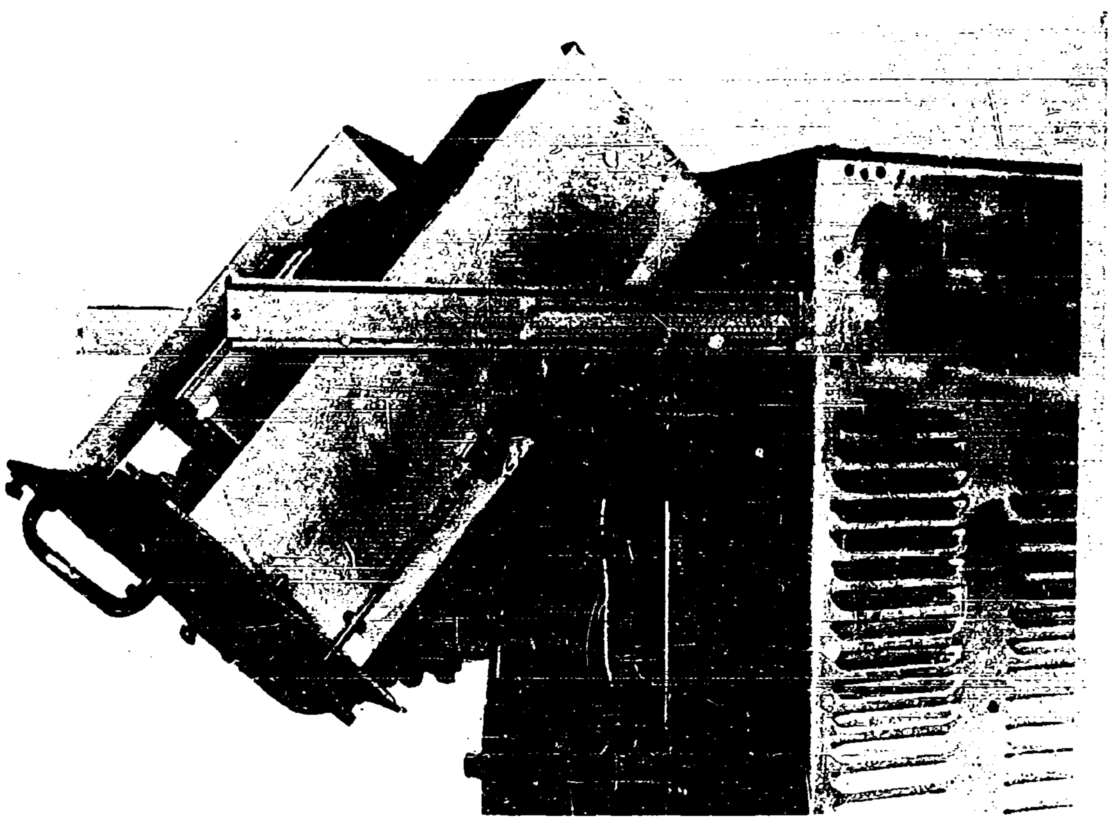


Fig. 33 - Chassis secured to ball-bearing slides with tilting feature for improved accessibility

Interchassis Cabling

The multiplicity of leads in interchassis cabling can be handled effectively by disconnect plugs or by service loops. Unless excessive amounts of relative motion take place between the cabinet and the rear of the chassis, quick-disconnect plugs seldom cause difficulty during shock and vibration, and they offer the advantage of permitting complete removal of the chassis from the cabinet for inspection or repair. Perhaps the frequently used combination of service loop and disconnect plug is the best solution to the cabling problem from a shock and vibration viewpoint, since "in position" servicing can be accomplished without an additional test cable, yet complete removal of the chassis is still possible. Where a service loop is necessary, firm clamping should be provided at the last point of contact on the chassis and the first point of contact on the cabinet. When the chassis is pushed into "locked-up" position, the loop should coil into place aided by a loop, spring, or the stiffness of the cable itself.

Corner and Flange Forming

The sequence of illustrations in Fig. 34 depicts one effective method of forming chassis corners. By cutting a 1/2-in.-wide slot from the original plate, as illustrated, a corner having excellent stiffness results. All bending radii should be not less than the thickness of the plate, to reduce the effects of stress concentration. Forming corners in this manner causes a loop to occur at the point of intersection of the three mutually perpendicular planes, which helps to minimize stress concentration. The lapped surfaces can be joined by rivets or spotwelds. This type of fabrication also provides an area of greater stiffness for supports such as guide pins. For small chassis, these receptacles can be simple steel plates drilled to produce a snug fit for the guide pins. For larger chassis, a steel sleeve can be welded to this plate to lower the bearing surface stresses. Some manufacturers incorporate a small 1/2-in. supplemental flange along the lower lip of the large vertical flange, as in Fig. 35. Because of interferences, the upper right portion of the flange on this chassis had to be removed. For the larger chassis, supplemental flanging is good practice.

Chassis Stiffness

As chassis become larger and loads become greater, the vertical natural frequencies are lowered. These vertical resonances can be raised by good component location and, if necessary, extra stiffeners. Figure 36 shows a chassis on which the parts were well arranged for attainment of a high vertical resonance. All transformers were located around the sides and rear of the chassis, and the lighter parts were centrally located. Shifting the center of gravity of the entire assembly rearward reduced the severity of rocking frequencies.

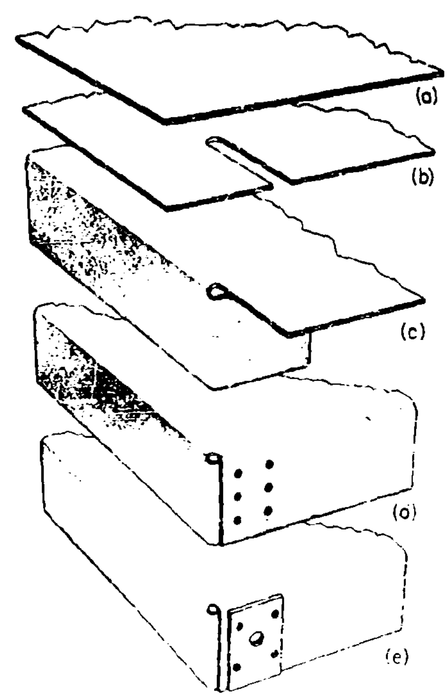


Fig. 34 - A method of forming chassis which produces stiff corners

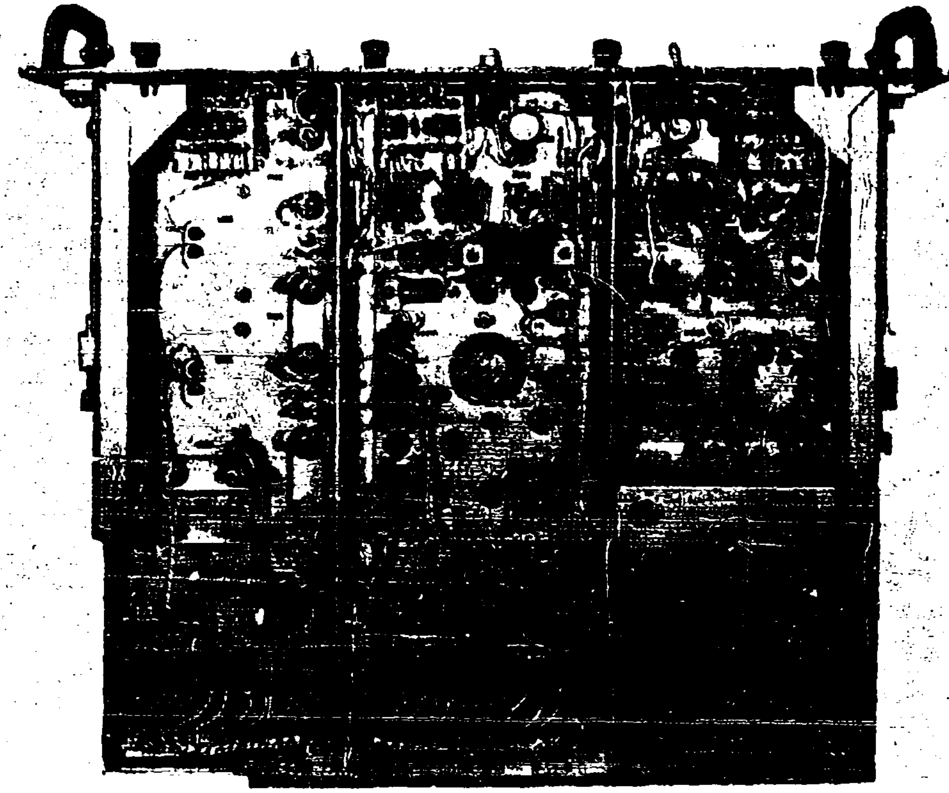


Fig. 35 - Chassis with additional horizontal flanges

No simple, accurate rule involving weight and size has been established to determine whether extra stiffeners are required in chassis. Reference 11 shows a method for the approximation of the values of vertical chassis resonances, and in the following section a sample calculation is carried out. If the approximated resonance falls below 35 cps, extra stiffeners are needed. For the average chassis, which support power transformers as well as the usual variety of vacuum tubes, condensers, and other miscellaneous components, it is recommended that extra stiffeners be incorporated into the design when the dimensions exceed 12 in. by 12 in. These stiffeners can be angles, channels, or other standard structural shapes. Usually a single stiffener across the center of the chassis is sufficient. In the case of the chassis shown in Fig. 35 (whose dimensions were approximately 23 in. by 23 in.), two angles were necessary.

Other possibilities for stiff chassis are shown in Figs. 12 and 37. In Fig. 12, unitized chassis construction is used, and such construction is popular where unhampered accessibility and quick replacement requirements are important. Because of the reduced sizes of the chassis, their individual resonances are quite high. The type of bracing shown in the unit in Fig. 37 although not common, does result in a stiff structure.

Resonant Frequency Determination

It is often difficult to decide, in the design stages, whether extra stiffeners are required to raise the vertical resonance of a chassis above the test frequency range, and

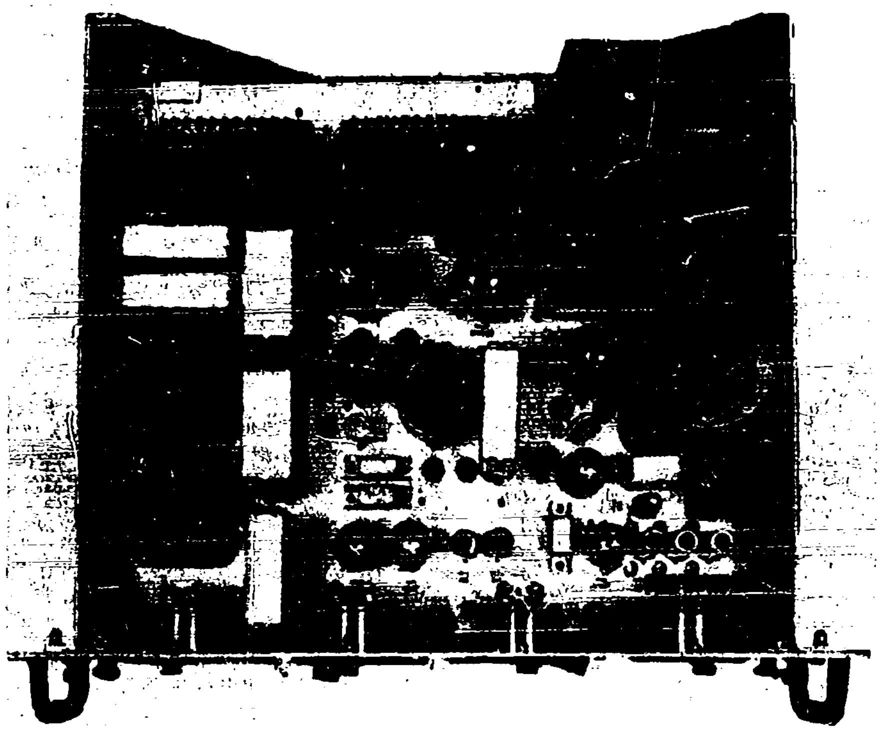


Fig. 36 - Chassis with good parts arrangement from a weight viewpoint

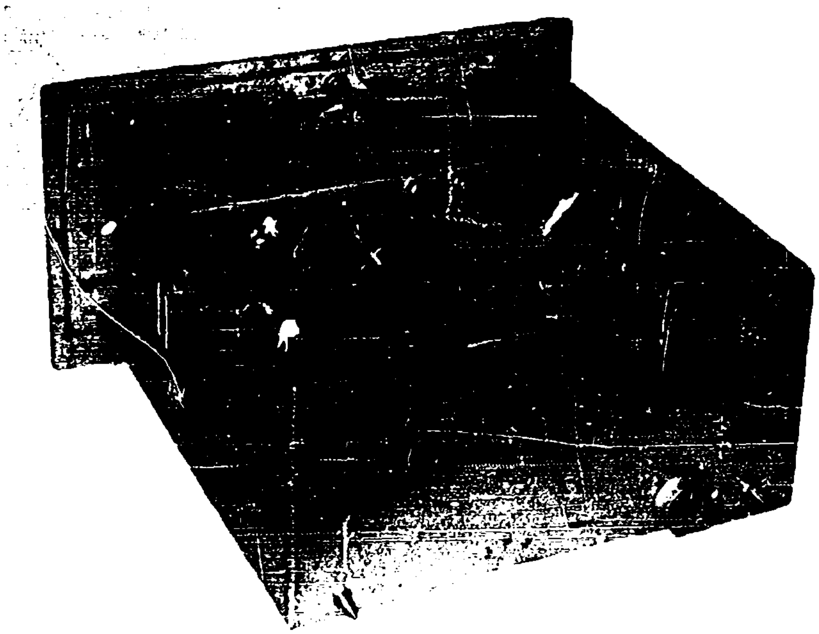


Fig. 37 - Deep chassis construction with vertical bracing or dividers

stiffeners are difficult to install after component mounting and wiring has been completed. It was suggested previously that chassis larger than 12 in. by 12 in. have stiffeners included to preclude any possibility of a low resonance. Actually, chassis somewhat larger than this might be satisfactory without extra stiffeners, and some smaller, heavily loaded chassis might cause difficulty. For this reason, the method presented in Ref. 11 for the determination of the resonance of a loaded chassis is included in this section. It is pointed out, however, that this method does not take into account the effect of holes or openings in the chassis, nor the possible stiffening effect of rigid components, such as transformers.

The study of vibrations of thin plates forms the basis for the analytical work. Since real chassis cannot be represented by either a simply-supported or clamped-edge plate, but actually by something intermediate, an exact resonant frequency determination for the first mode of vibration of a loaded chassis cannot be made. However, two frequencies, one for a loaded plate with simply-supported edges and the other for one with clamped edges, can be determined, and these values form boundaries for the actual resonant condition of the chassis. It was found that smaller chassis have resonances which fall nearer to the clamped-edge condition, whereas the resonances of the larger chassis approach those of a simply-supported plate.

The problem of calculating resonance is further complicated by the fact that the average chassis has a number of weights mounted on it in random locations. Those weights located away from the center have less effect on the resulting natural frequency than equal weights mounted at the center. A weight located at the center of the chassis that produces the same effect on the natural frequency as another weight located away from the center is called the "equivalent center weight." Therefore, by adding "equivalent center weight," a "total equivalent center weight" is found which should affect the resonant frequency of the chassis the same as that of all the individual weights.

In Figs. 38, 39, and 40, "equivalent center weight" factors from Ref. 11 are presented for three sizes of aluminum chassis. It was found that the analytical determination of the "equivalent center weights" was not in very close agreement with experimental results, since the exact deflected shape of the plate for the first mode was not used in the solution of the energy equations from which this method was derived. The curves in the figures were obtained by actually vibrating chassis with weights mounted on them. In calculating the resonant frequency of a chassis of a size different from those for which curves are given, the set of curves corresponding to the nearest size chassis should be used. It is, of course, possible to determine experimentally curves for a particular chassis size, if greater accuracy is demanded.

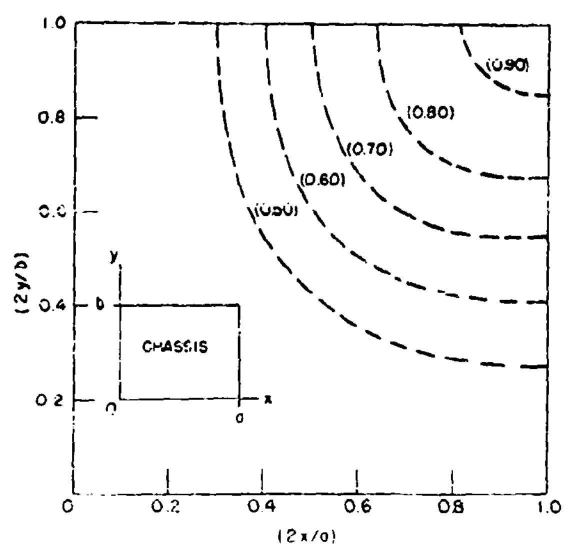


Fig. 38 - Plot giving "equivalent center weight" factors for an aluminum chassis whose dimensions are 7 x 9 x 2 in.

Figure 41, from Ref. 11, presents curves for a simply-supported and clamped-edge plate, relating the actual resonant frequency of the loaded chassis, the fundamental resonance for the unloaded chassis, the "total equivalent center weight," and the weight of the plate (or chassis, excluding flanges). Between

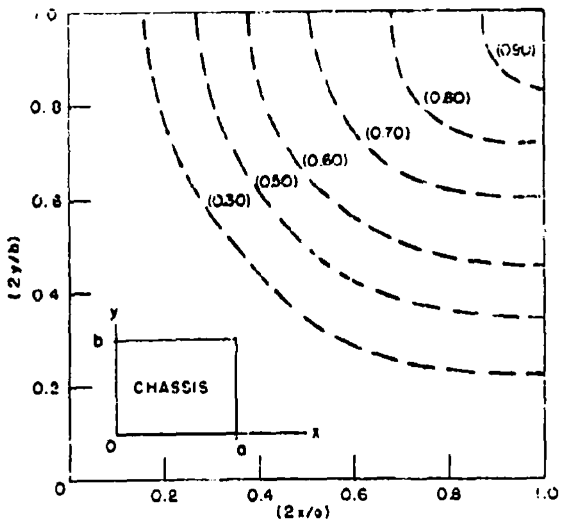


Fig. 39 - Plot giving "equivalent center weight" factors for an aluminum chassis whose dimensions are 10 x 12 x 3 in.

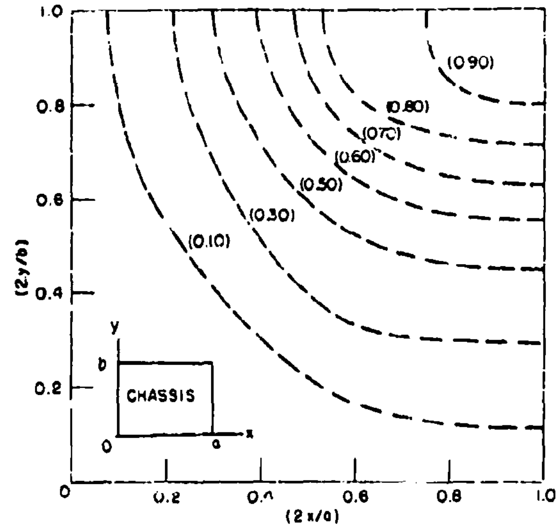


Fig. 40 - Plot giving "equivalent center weight" factors for an aluminum chassis whose dimensions are 13 x 17 x 3 in.

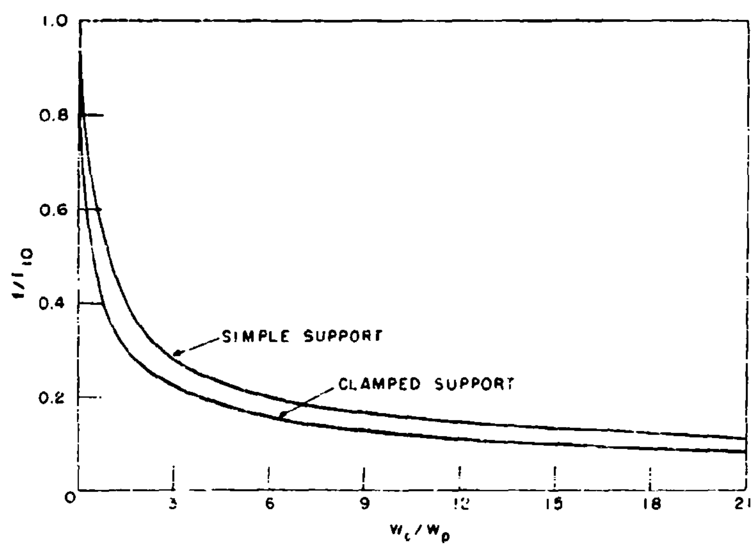


Fig. 41 - Curves relating W_c/W_p (Eq. 2) and f/f_{01} (Eqs. 3 and 4) for simply-supported and clamped-edge plates

these curves will lie the resonance of the actual chassis. The coordinates of Fig. 41 are defined by the following equations:

$$\frac{W_c}{W_p} = \frac{W_c}{\rho g a b h} \tag{2}$$

$$\frac{f}{f_{01}} = \frac{f}{h\phi\left(\frac{1}{a^2} + \frac{1}{b^2}\right)} \quad \text{(simply-supported condition)} \tag{3}$$

$$\frac{f}{f_{01}} = \frac{f}{h\phi_1\sqrt{\left(\frac{3}{a^4} + \frac{2}{a^2b^2} + \frac{3}{b^4}\right)}} \quad \text{(clamped-edge condition),} \tag{4}$$

where

W_c = equivalent center weight, pounds

W_p = weight of plate = $M_p g$, pounds

ρ = density, slugs/cu in.

g = acceleration of gravity, in/sec^2

a, b = lateral dimensions of plate, inches

h = thickness of plate, inches

f = frequency, cps

f_{01} = fundamental frequency of system with no weights attached, cps

ϕ = 133,000 for aluminum

= 174,000 for steel

ϕ_1 = 177,000 for aluminum

= 232,000 for steel.

Reference 11 describes, in its illustrative example, a procedure for finding the required thickness of a plate chassis such that, for a particular distribution of weights, the resulting resonance will be above some specific frequency. An illustration of a resonant frequency approximation by this method is given here, since one has little choice, practically, to vary the plate thickness.

For illustration, the resonance of an aluminum chassis 17 in. long, 13 in. wide, and 3 in. deep, weighing 40-1/2 lb, was calculated, and the chassis was later vibrated to check the accuracy of the calculation. The chassis was formed of sheet material 0.064 in. thick. On the chassis were located four transformers and one condenser. Since this system was set up specifically for the purpose of illustration, no wiring or small parts were included, in order to decrease the work of calculation. Although wiring is not considered in the approximation, it should have little lowering effect on the resulting resonant condition.

Location and weight of each component was found and tabulated. The x- and y-axes were respectively along the long and short dimensions of the chassis. Since Figs. 38, 39, and 40 represent a quadrant with the origin of the coordinate system occurring at the outside corner, the x and y coordinates of the masses were measured from the outside edges of the chassis and were subsequently modified by letting $X = 2x/a$ and $Y = 2y/b$. "Equivalent center weight" factors, corresponding to X and Y for each individual component, were then secured from Fig. 40, the figure whose chassis dimensions correspond to those of the given chassis. By multiplying the weight of each individual component by its related factor, the "equivalent center weight" was obtained. A chart similar to the following can be set up to clarify and expedite the calculation.

Component	Wt. (lb)	x	y	X	Y	Factor from Fig. 35	ECW
Transformer T1	12.25	3.12	3.06	0.37	0.47	0.22	2.70
Transformer T2	12.25	3.12	3.06	0.37	0.47	0.22	2.70
Transformer T3	7.63	3.50	3.50	0.41	0.54	0.33	2.52
Transformer T4	4.88	3.50	3.00	0.41	0.46	0.27	1.32
Condenser C1	0.69	8.50	6.50	1.00	1.00	1.00	0.69
						Total	9.93 lb

Substituting the total ECW in Eq. 2 (note: $\rho g = 0.1$ lb/in^3),

$$\frac{W_c}{W_p} = \frac{9.93}{(0.1)(13)(17)(0.064)} = \frac{9.93}{1.42} = 7.00.$$

Corresponding to this value of W_c/W_p, values of f/f_{01} can be obtained from Fig. 41 for clamped- and simply-supported conditions. These values are:

Simply supported: $f/f_{01} = 0.18$

Clamped: $f/f_{01} = 0.14$

By substituting in Eqs. (3) and (4), resonances corresponding to each condition of support can now be determined.

For the simply-supported condition:

$$f = (0.18)(133,000)(1/13^2+1/17^2)(0.064) = (0.18)(133,000)(0.009)(0.064) = 13.8 \text{ cps}.$$

For the clamped-edge condition:

$$f = (0.14)(177,000)(0.064)\sqrt{\left\{\frac{3}{17^4} + \frac{2}{17^2\,13^2} + \frac{3}{13^4}\right\}} = (0.14)(177,000)(0.064)(0.014) = 22 \text{ cps}.$$

Consequently, resonance of the first mode must fall between 13.8 cps and 22.2 cps, the average of which is 18.0 cps. By vibrating the chassis, resonance was located at 19.3 cps. Hence the calculation was sufficiently accurate to indicate the need for additional stiffeners. For this particular chassis, a single 1-in. aluminum angle, or equivalent, fastened at the center of the chassis along the shorter horizontal dimension, would have been adequate to raise resonance above maximum testing frequencies.

The same calculations were carried out with a steel chassis. All other aspects of the problem remained the same (weights, locations, etc.), except the plate thickness, which was reduced to 0.047 in. Using Figs. 40 and 41, the average of the two calculated frequencies was found to be 25.6 cps. Actual vibration showed resonance to be at 22.5 cps. Considering the nature of the problem, and the fact that Fig. 40 was based on an aluminum chassis, the designer would have expected a lower resonant condition. Because of the divergence between actual and calculated frequencies using this method, 35 cps should be considered the lower design limit. Chassis stiffeners should be introduced if the calculated frequency falls below 35 cps. Resonance of the higher modes would be above these frequencies.

CATHODE-RAY TUBES

Introduction

The protection of cathode-ray tubes from shock and vibration presents a difficult problem. Even for normal handling operations, hazards exist which are greatly magnified when high-impact shock is introduced into the environment. The importance of proper support and mounting of these tubes cannot, therefore, be too highly stressed.

Typical Damages

Glass breakage is the most common type of CR tube shock damage. Breaks are usually confined to the neck of the tube, where flaring toward the face begins. In several installations, the internal elements of tubes were permanently deformed during shock, which resulted in permanent misalignment of the electron beam, although the tube itself still operated electrically. Where friction provides the only force positioning the tube in its cradle, rotation under shock sometimes takes place, causing the scope presentation to become canted. Sockets that are improperly restrained provoke electrical maloperation. Under vibration, about the only type of damage normally is tube element failure.

Mounting Practices

One good method for mounting CR tubes is demonstrated in the mounting of the 5-in. tube shown in Fig. 42. Around the envelope of the tube is a formed sheet-metal housing. The housing serves two purposes. From the electrical viewpoint, it protects against stray magnetic and electrostatic fields; in addition, it provides an excellent means of mechanical support. Between the tube and the shield, firm rubber padding (not sponge rubber) is placed to prevent any buildup of a concentrated loading, and to reduce the intensity of the level of shock.

Stiffness in the supporting structure is important. If excessive flexibility exists, the neck of the tube can easily be snapped during shock by twisting or bending. In this particular construction (Fig. 42), aluminum angles formed the supporting superstructure and performed satisfactorily in both shock and vibration.

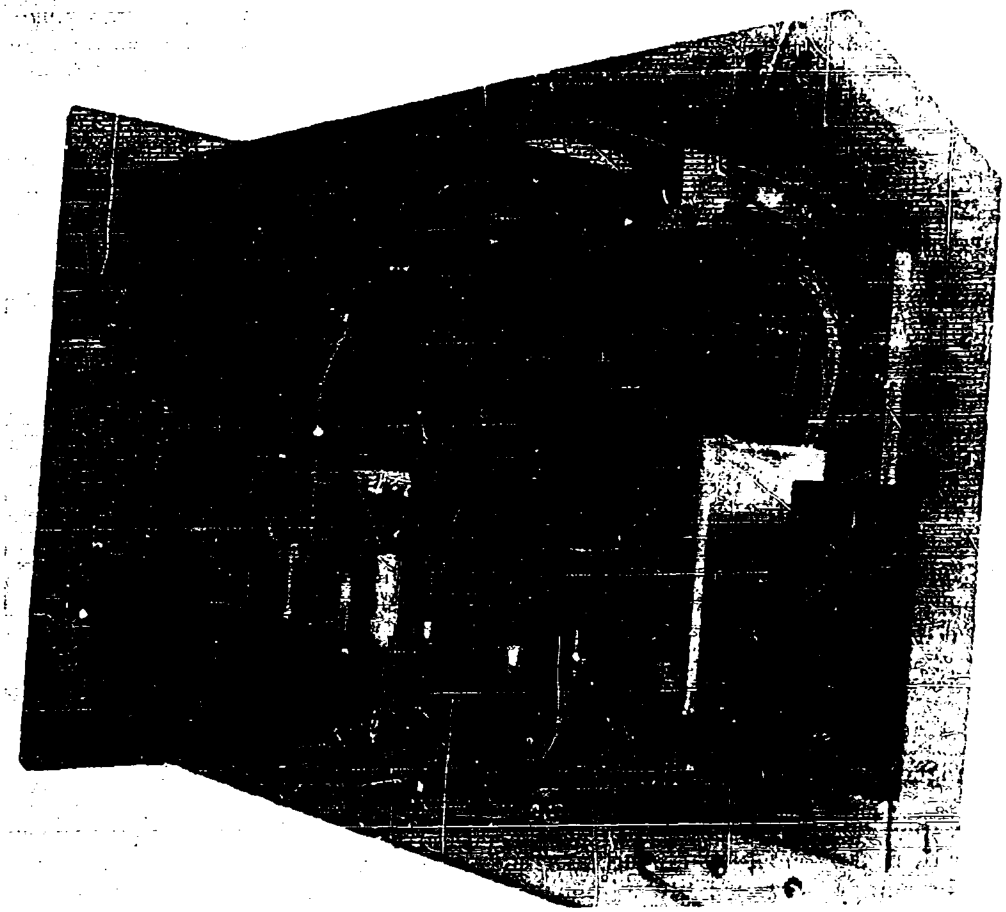


Fig. 42 - A good mounting for a 5-in. cathode-ray tube

The tube is inserted neck first into the housing and is held in place by a molded rubber bezel fitting around the outer periphery of the face. A suitably shaped metal retaining ring is bolted over the bezel. The socket of the tube is spring-loaded to force the tube against the rubber bezel, or face-piece, to take up any looseness in the mounting. Caution should be exercised to limit the deflection of the spring-loaded assembly under longitudinal shock. If large tube excursions are permitted, the resulting rebound impact between the tube and bezel might possibly cause damage. Therefore, by the use of such a design technique, any motion of the tube, regardless of the direction of shock, is arrested by rubber padding.

The housing is supported at both ends. In the front, a circular flange is spotwelded to the housing and bolted to the front panel, and in the rear another spotwelded flange is fastened to a metal overhead hangar. Passing through the rear flange are two rods which attach to a rear plate. The rear plate fits freely over the end of the socket and prevents the socket from being unseated during shock. A second rear plate, just forward from the first, is attached rigidly to the socket. This plate and the tube can be rotated independently from the rest of the mounting assembly. Rotation of the forward plate provides a means for scope presentation alignment. When both plates are locked together, the tube cannot rotate.

The problem of support becomes more critical with increased size of the glass envelope. The same techniques as previously illustrated and discussed can be used for the larger tubes, but definite precautions must be taken to insure adequate stiffness in the mounting.

Radar viewing scopes frequently run as high as 21 in. in diameter. For these larger tubes often mounted vertically, the shock motions must also be arrested by padding between the surfaces of the envelope and the housing in order to distribute the resulting inertial loads over as large an area as possible.

Instead of using springs to eliminate looseness in the mounting, adjustable rubber snubbers, fitting around the socket of the tube, would be a better substitute. The snubbers would be designed to act on plates fastened over the socket, as shown in Fig. 42.

Concluding Comments

The CR tube must be floated in rubber of adequate firmness. Rubber not only absorbs part of the shock, but it accommodates some twisting of the supporting structure without damage to the tube, and protects the brittle glass envelope from concentrated loadings. Suitable restraining devices have to be provided for the socket. Since ruggedness increases as envelope size decreases, it is best to use the smallest possible tube that fulfills operational requirements.

PARTS

Electromagnetic Relays

Relays present two principal problems. First, there is the problem of balance associated with the armature-contact assembly. Dynamic balancing of the assembly of the smaller relays is reasonably common practice, but less common on the larger. The second problem is closely allied with high-sensitivity requirements in modern high-speed equipment. As the need for greater sensitivity increases, air-gap clearances become smaller, and forces used to maintain armature positions become less.

A sensitive balanced-armature relay is shown in Fig. 43. Vibrationwise, the relay's performance was satisfactory for shipboard conditions. It was stated that the relay could withstand vibrations of 10 g amplitude for frequencies not greater than 60 cps. This is more severe than the actual test conditions, up to 2 g at frequencies of 25 cps and below. Shipboard vibration does not present a severe relay problem. But shock, on the other hand, presents serious difficulties. For example, the relay of Fig. 43 was subjected to a shock on the Navy lightweight, high-impact, shock machine, and was monitored by an oscilloscope whose trace indicated the motion or bouncing of the movable contacts between the fixed contacts. A high-speed camera recorded the scope presentation. To provide a time history of the relay's performance, a 500-cps timing trace was also included.

Figure 44, a sample film strip showing the performance of the relay under shock, graphically portrays the effect of shock-excited vibrations of the armature. The sinusoidal pattern on the strip is the timing trace, on which time increases from left to right. Above the timing trace is the trace showing the positions assumed by the movable contacts during shock. The solid line at the lower end of the trace was recorded with the armature in the energized position before the blow was struck. Power was maintained on the coil throughout the disturbance. Disturbance of this trace indicates the time at which the blow was struck. The lower, middle, and upper concentration of lines during the disturbance is the position taken by the electron sweep corresponding to the energized, neutral, and de-energized positions of the armature. According to the time trace, the armature disturbance lasted for approximately 335 milliseconds with single contact openings of up to 3.5 milliseconds duration.

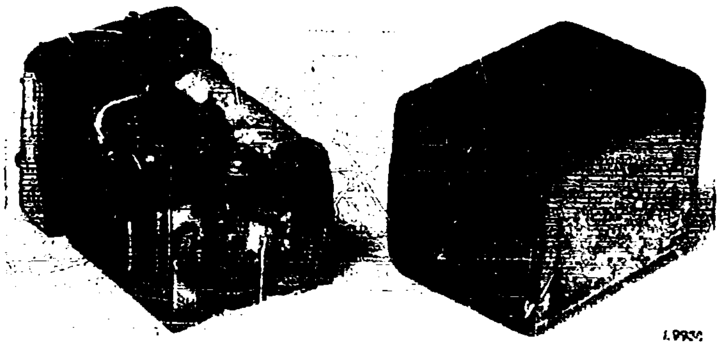


Fig. 43 - A sensitive balanced-armature relay

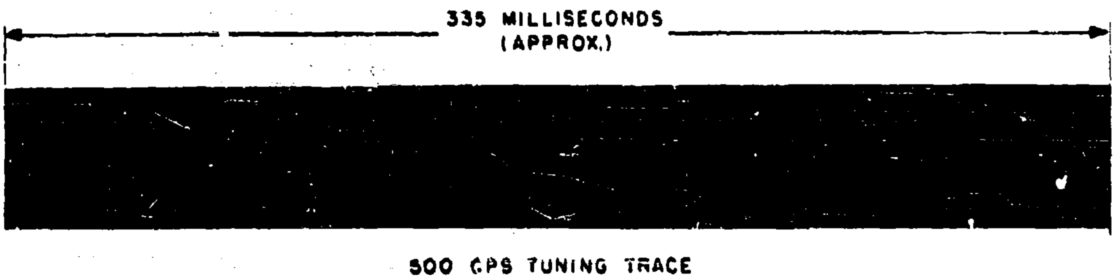


Fig. 44 - Oscilloscope photographic record of momentary openings and closings of the movable contacts of a relay under high-impact shock

Performance of the relays under shock and vibration is usually better in the energized condition, because the magnetic field of the coil tends to damp or restrain the motion of the armature. For example, in the case of the relay described here, the duration of the shock disturbance when energized was in one case only 30 percent of the duration of the disturbance in the de-energized condition. Performance in this regard is greatly affected by the direction of the shock disturbance. A horizontal shock parallel to a plane through the axis or pivot points of a relay armature usually has the least disturbing effect. Advantage should be taken of these facts in the circuit design and in the physical orientation of the relay whenever possible.

Relay manufacturers have developed some interesting designs in their search for shock resistance. Figure 45 shows a rotary-type relay in which the armature tends to align itself with the magnetic field of the energized coil. Unhappily, in this instance, the same shock problems existed, with performance no better than that of the previous type.

Although no relay has as yet been designed that possesses positive operational characteristics under both shipboard vibration and shock, it does not mean that relays cannot be used successfully under such conditions. Circuits containing relays can be so designed that automatic recycling and restarting occurs after each shock, or, as is often the case, circuit response can be made slower than the shock disturbance, with the result that malperformance does not have a chance to take place.

Fig. 45 - Rotary-type relay

Because of the importance of the relay in electronics, much research is being directed toward it and its operation. The approaches that show most promise for better shock resistance are the development of very light and rigid armature sections, increased forces for holding armature in position, dynamically balanced elements, wiping and locking type contacts where permissible, and adjustment of circuit response times to maximum values.

Capacitors

The capacitors most frequently used in electronic equipments may be grouped, according to their general methods of construction and mounting, into three basic classes. These are the lead-mounted types, which assume many varied shapes and sizes, and which have supplemental support provided for the heavier ones in the form of a body clamp, the canned types, in whose design the provisions for mounting may vary widely, and the variable air-gap types. By virtue of their basic design, classes one and three present the fewest problems of shock and vibration.

Damage can also be grouped into three classes, which are damage to internal leads as in the case of the canned types, damage to external leads, and damage to or inadequacy of the mounting or clamping system. These classes are not necessarily independent of each other, yet each may occur without the other two resulting. For example, it is possible to mount a canned condenser so flexibly that as a consequence both internal and external lead failures occur. Conversely, internal leads may fatigue because of improperly supported elements within the condenser housing, without the condenser itself being too flexibly mounted.

To solve the problem, the designer must first choose components in which all elements have good support. By this is meant that all resonant frequencies must be well above anticipated frequencies of the disturbing vibration. Obviously, the overall mounting of the condenser must possess the same characteristic. Capacitors are rugged and are seldom damaged. Most of the damage that does occur can be attributed to poor mountings.

Figure 46 shows typical damage to a canned bathtub capacitor. Relatively few cycles of vibration were required to cause fatigue of the mounting ears. The chassis on which this was mounted also possessed a high degree of flexibility, which contributed greatly to the failure. However, chassis flexibility is not always correctable if excessive,

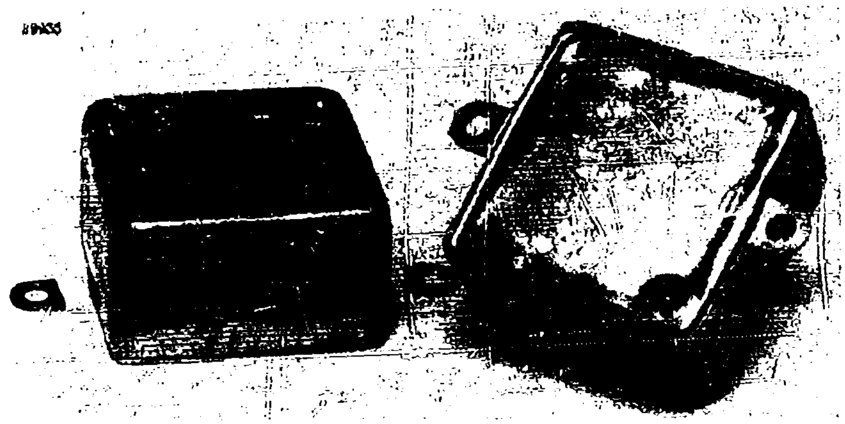


Fig. 46 - Typical damage to a canned bathtub capacitor

and components having little vibration resistance should not be used. In the case of the damaged capacitor, a metal retaining strap over the housing was necessary until structural modifications could be made to stiffen the chassis.

Broken leads on lead-mounted capacitors can usually be prevented by a metal clamp. The same rules given for small resistor (see page 50) that are lead-mounted apply to lead-mounted capacitors. Excessive flexibility and lead failures in canned and air-gap types can be prevented most easily by substitution of more rugged types which are normally available.

The types of capacitors and mountings shown in Fig. 47 are usually satisfactory, whereas those in Fig. 48 are somewhat less desirable, especially for the larger capacitors of their particular series. Under shock, condensers held by fuze clips, as in Fig. 48a, sometimes become unseated because the spring pressure is not adequate. Under vibration, occasional fracture of the solder bonds on the twist-lug types, Fig. 48c, are a source of malperformance because of the resulting looseness to the mounting and disruption of circuit continuity. Soft solder is also used to bond the mounting bracket to the body of the type shown on Fig. 48b, thus making this design somewhat susceptible to fatigue. However, all of the designs in both figures are used successfully for the smaller physical sizes. Most commercial capacitors, when mounted properly, possess sufficient vibration and shock resistance to meet the requirements of current Navy test specifications.

Transformers and Chokes

Lack of stiffness is the direct cause of most transformer damage. In laboratory evaluation tests, more transformer damage is caused by vibration than by shock. Figures 49 through 52 illustrate some of the more common small transformer designs used in current electronic equipments. The coil and core of the transformer in Fig. 49 is supported in cantilever fashion, which results in low resonant frequencies. This type of mounting appears much too frequently. A better arrangement is to mount the core and coil on symmetrical supports, as shown in Fig. 50. By using the same bolts which tie the core to the housing as the chassis mounting bolts, the housing is not subjected to large concentrated loads. As a result, a thin sheet-metal housing is adequate, as it serves simply as a reservoir for the cooling or potting compounds. Because the housing of the transformer in Fig. 51 was required to bear the weight of the entire transformer assembly, the mounting flanges fractured under vibration, and internal leads failed.

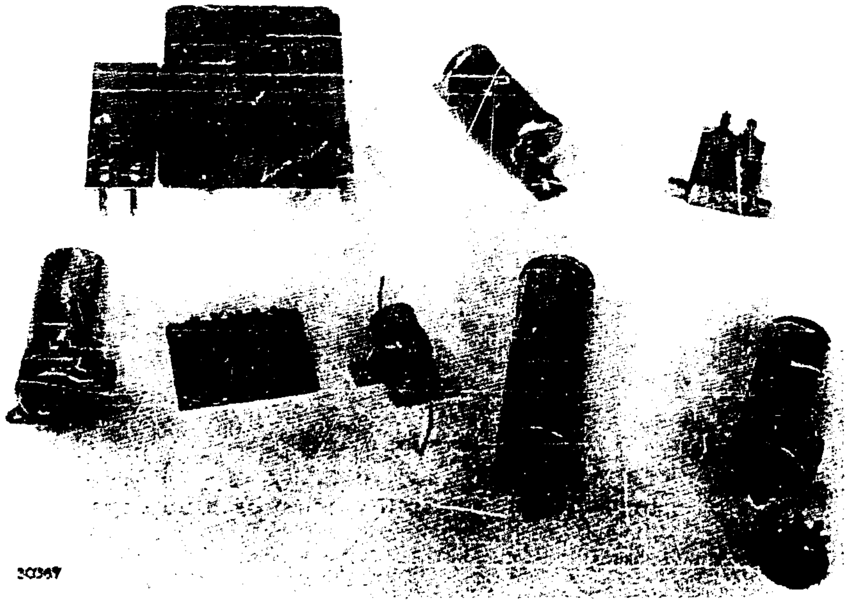


Fig. 47 - Types of capacitors and mountings usually satisfactory for shock and vibration environment

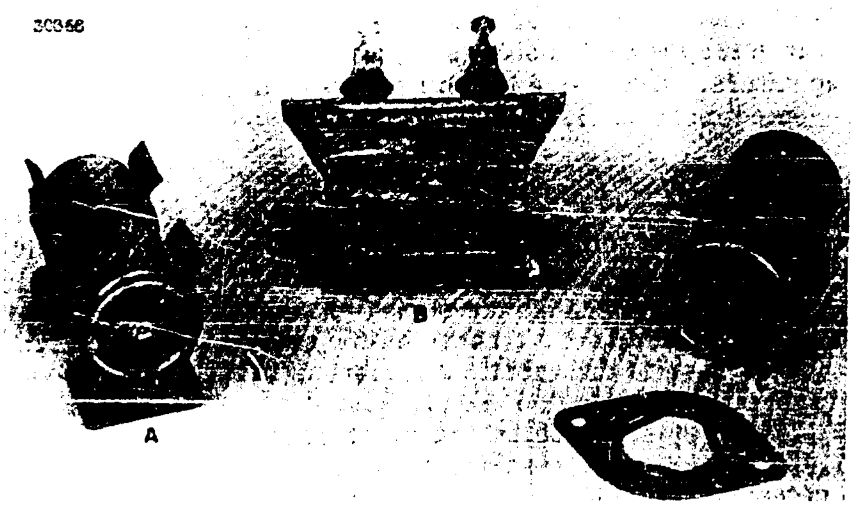


Fig. 48 - Types of capacitors and mountings less desirable for shock and vibration environment, especially for the larger sizes

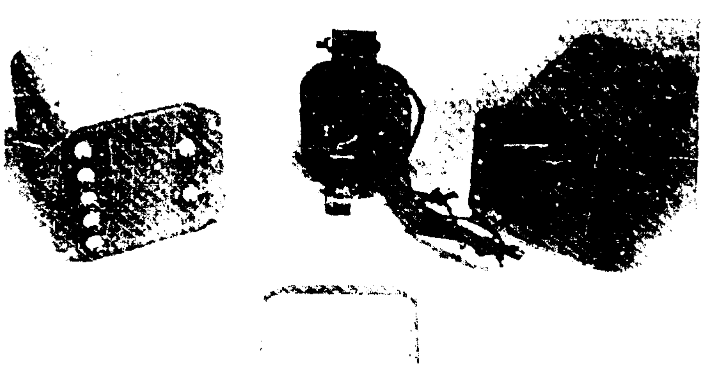


Fig. 49 - Cantilever support for transformer coil and core which results in low resonant frequencies

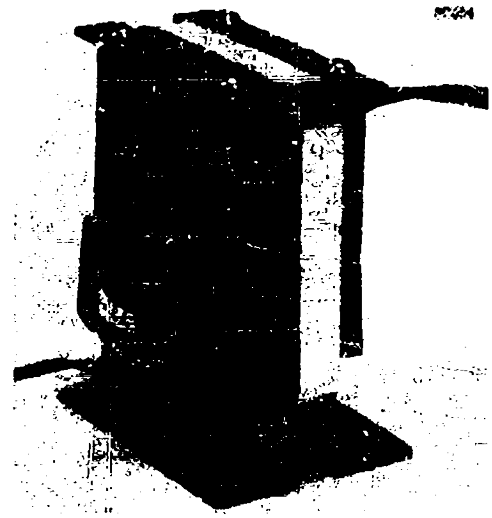


Fig. 50 - Transformer core and coil mounted on symmetrical z-section supports

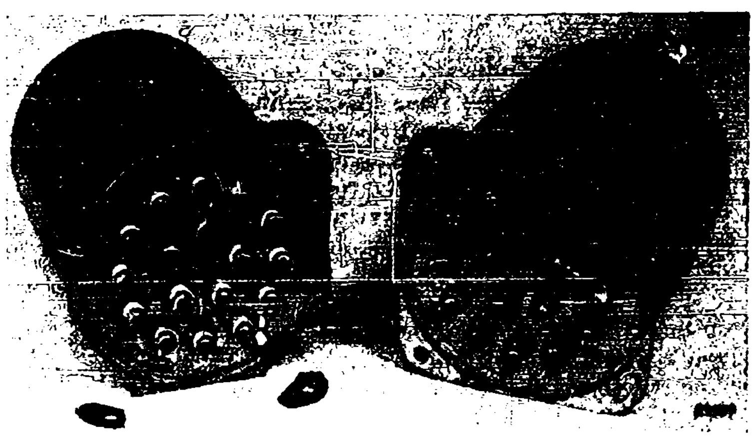


Fig. 51 - Fractured mounting flange of transformer housing supporting entire weight of assembly

Fig 52 - Desirable transformer core mounting for shock and vibration environment showing mounting bolts passing through the core

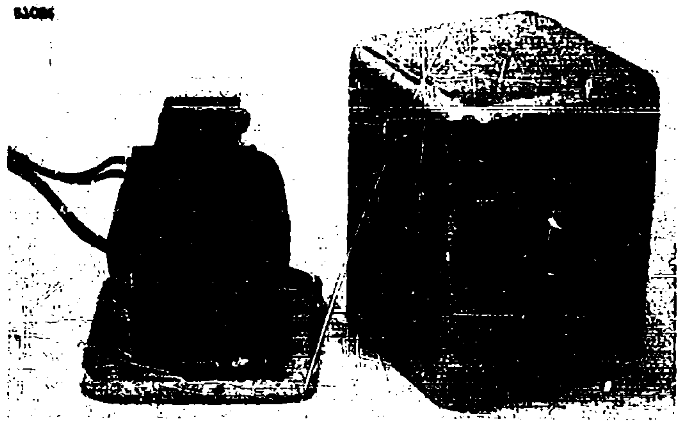


Fig. 53 - Transformer mounting showing core secured to bottom angle brackets

A desirable core mounting, from a shock and vibration viewpoint, is similar to that in Fig. 52, where the mounting bolts for the transformer pass through the core itself. Since "C" type cores (Hipersil, etc.) cannot be drilled without alternating electrical characteristics, adequate mountings become a more difficult problem. Figure 53 demonstrates the more conventional mounting method, where the core is bolted to angle brackets. Usually, no mechanical support is provided at the top of the core. A stiffer system could be effected by similarly supporting both the top and bottom of the core, using the housing as a load bearing structure.

An influence is also exerted on the stiffness of any core mounting by the potting components used. Pitch, having a much greater viscosity than oil, increases resonance and provides better support, but other characteristics, such as dielectric strengths and heat conductivities, might prohibit its use in many designs. Compounds of the highest viscosity compatible with all design considerations are preferable.

Transformers and chokes are a fruitful source of substantial reductions in weight of electronic equipments. Their miniaturization by the use of better designs and materials would go far to reduce the shock and vibration problems of electronic equipment.

Resistors

Resistors perform exceptionally well under shipboard shock and vibration, when mounting practices conforming to such environmental conditions are stringently followed. Occasional damages that do occur are not the fault of the resistor itself, but of the method and manner of mounting.

Resistor mountings are very similar to those of capacitors, which have been described in another section. Greater emphasis should be placed on adequate support for the phenolic boards commonly used to mount resistors of the 1/3- to 2-watt ratings. Too often, the distance between supports of these mounting boards cause low-frequency resonance to result. This in turn causes lead failures.

Before solder is applied to any terminal connection, a good mechanical connection should exist. This is obtained by wrapping the end of the lead securely around the point of connection. Some manufacturers allow approximately 1/8 in. of free lead to protrude

from the wrapping after trimming, to make replacement easier. For increased stiffness, the lead length between the resistor body and the binding post should be maintained at a minimum, probably no more than 1/4 to 3/8 in. for the 1/3- to 2-watt sizes. No damages result when these few simple practices are followed.

Potentiometers are also resistant to damage, although rotation may occur under vibration and shock conditions if locks are not provided. Commercial locks are satisfactory for shipboard vibration, and should be used.

A word of caution concerning body clamps: leads break if clamps are not used for lead-mounted resistors larger than the 2-watt size. The solution is to clamp any part where doubt exists as to its resonance.

Air Blowers

The large amounts of heat energy released in electronic equipments during operation make it necessary to provide cooling to prevent unsafe temperature elevations within the enclosures. To control these enclosure temperatures, propeller or centrifugal-type blowers, as shown in Figs. 31 and 54, respectively, are frequently used.

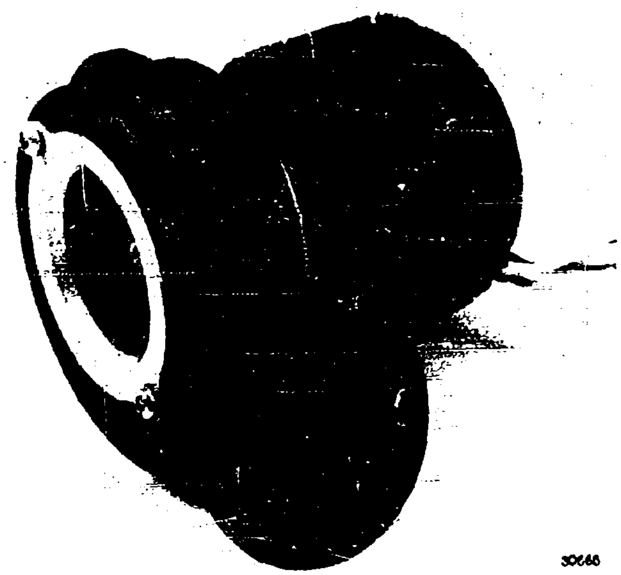


Fig. 54 - Poor mounting arrangement for a centrifugal-type blower

As in the case of many other components, the blowers themselves, if solidly mounted to the structure, possess in general a sufficiently high degree of ruggedness to withstand shock and vibration. The methods of mounting create the weak link in the chain. Such deficiencies are intensified by the lack of mount symmetry and the use of mounts to provide vibration isolation, where noise elimination is necessary. The blowers should not be provided with isolation mountings, unless the generated microphonics are of high enough magnitude to cause difficulty. If they are used, sufficient clearance and damping must be provided. From the viewpoint of the discussion, it is more satisfactory to mount the blower solidly in the equipment.

Figure 54 illustrates a poor design of centrifugal-type blower for vibration and shock resistance. The weight of the entire assembly is supported by the flange surrounding the outer end of the delivery duct. This results in a cantilevered system. This type of mounting is unfortunately quite prevalent in many equipments. The solution is to provide additional support for the motor. Where the motor housing has tapped holes which can be used to receive supplemental fasteners, support can easily be provided. But in the few cases where such holes do not exist and cannot be made, strapping becomes the simplest expedient. Metal straps passing around the motor body and fastened to suitable brackets as a rule introduce adequate stiffness into the assembly.

Propeller-type fans present less of a problem, since adequate motor support is normally furnished. If it is not, corrective modifications should be made.

In summary, vibrational resonances resulting from poor mountings are the chief reason for damage to blowers, according to laboratory tests. Bearings, end bells, armature assemblies, etc., all seem to possess the necessary stiffness and strength requirements, when dynamic conditions are not unnecessarily amplified by resonances.

Fasteners

Recent years have witnessed a tremendous increase in the complexity of equipment, and along with this increase, the need for rapid inspection and repair techniques has acquired greater importance. To meet this need, quick-release-type fasteners have come into widespread use, especially in the aircraft and electronics industries. Their desirability arises from the fact that only a fraction of a turn is necessary to lock or release the device; complex equipment, if joined together by such fasteners, can be disassembled, inspected or repaired, and reassembled in less time than is required for the same operations using screws and bolts. A few of the more popular types are illustrated in Fig. 55.

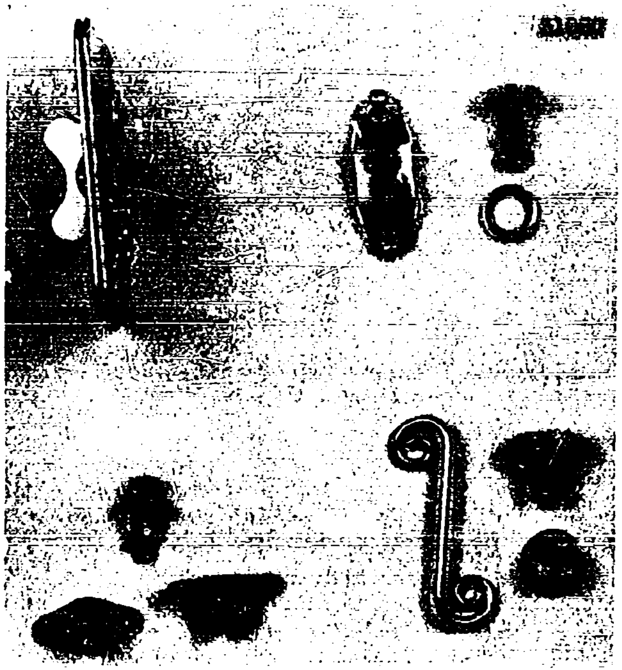


Fig. 55 - A few of the more popular types of quick-release fasteners

Quick-release fasteners are infrequently used in shipboard units, and then usually in locations not subjected to large loads. A few manufacturers have used them in light sub-chassis mountings, but, under shock and vibration tests, bolt substitutions were necessary in several cases. Where they are required to carry loads of any magnitude, their performance is poor. Under shipboard vibration, these fasteners tend to allow relative displacements between the captive load and support, which in effect increases the transmissibility and the possibility of damage. Their strength under high-impact shock is much less than similar size screws or bolts. Although designers can use them with success to secure inspection plates or covers, it is wiser to use standard bolts and screws for most purposes where parts or subassemblies must be replaceable or removable for servicing. The basic weakness of quick-release fasteners arises from the fact that they employ a spring element to maintain tension when "locked up." Inertial forces of low magnitude overcome this spring force and cause looseness in the assembly. Under high-impact shock, the spring, or the cam rider—the member which engages the spring and through which the loads are transmitted from the captive mass to the support—can easily deform or fracture. Fractional-turn locking, however, would be generally impractical without such features.

Snap-rings, set-screws, and roll-pins are frequently unsatisfactory for shipboard conditions. Under shock, snap-rings can be easily unseated, and roll-pins may collapse. Set-screws are usually relegated to the task of positioning the smaller control knobs. Spotwelding, although possessing great popularity, is another method requiring prudent judgement. The basic structure of an equipment should not be spotwelded; however, the metal skin, or covering fitting over the frame, and many lightweight appendages, can be successfully installed by this method.

Figure 56 illustrates the type of fastener discussed under cabinets and consoles in conjunction with chassis mountings (Fig. 25). These fasteners are distributed around the front panel of the chassis to secure the chassis to the main frame of the cabinet, and are knurled and slotted to facilitate loosening and tightening. Shank diameters vary, of course, according to various design parameters, but should not be less than 3/16 in. for lightweight or 5/16 in. for mediumweight equipment. Normal thread sizes vary from 1/4 in.-20 to 3/8 in.-16.

The best fasteners or fastening methods for shipboard conditions are usually the more conventional, i.e., rivets, screws and bolts, and welding. With screws and bolts, lock washers—preferably split-ring types, since under shock they resist surface deformations which could lead to looseness—must be universally employed. Strength calculations for bolts are dealt with in some detail in Ref. (12), together with discussions concerning riveting and welding.

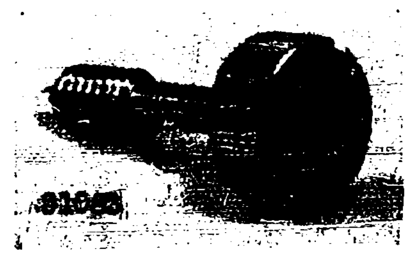


Fig. 56 - Reduced-shank front-panel thumbscrew

Electron Tubes

Tube performance is affected by many factors. Tube ruggedness, tube location on the chassis, chassis location in the equipment, and mounting methods, all influence overall performance. For example, greater shock protection is afforded when tubes are grouped near the center of chassis, where greater deflections occur. Tubes located distant from mounting surfaces benefit from attenuation which occurs as the shock passes through various elements of the structure. But, again, the primary causes of most tube damage during laboratory tests are resonant frequencies falling within the range of testing frequencies. Resonance of either the tubes themselves, or the chassis or structure on which the tubes are mounted, causes most detrimental results.

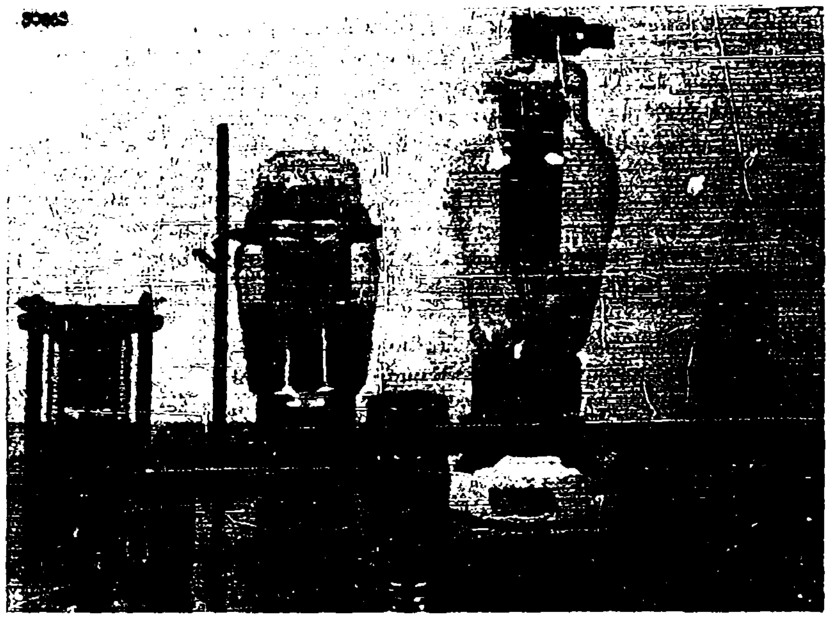


Fig. 57 - A few of the more popular types of tube clamps

Laboratory evaluation tests demonstrate that roughly twice as many tubes are damaged by vibration as by shock, for current shipboard acceptance specifications. The vibration part of a test is usually performed first, hence some apparent vibration failures might be attributable to the lack of quality control, but such failures are believed to be relatively few. The most important problem, therefore, is to raise the various individual resonances above predominating environmental frequencies. Other sections discuss the necessary measures required to stiffen, support, or form the chassis or structure.

Tube clamps are always necessary to retain the tubes in their sockets. Figure 57 shows a few of the more frequently used types in current electronic equipment. All of these are satisfactory, with proper application. Any of the clamps shown on the miniatures in the front row work equally well, as do the clamps used on the two outboard tubes in the back row. However, if extremely large tubes are clamped by devices similar to those shown on the two rear inboard tubes, a low resonance might occur. Better results could be obtained with two binding posts instead of a single one for the "top hat" clamp at the left, while a "top hat" used in conjunction with the bayonet base would be a worthwhile addition for the other system. These clamps will satisfy most needs. The tube manufacturer should be consulted to assure correct clamp application.

Another difficulty frequently encountered in shock is envelope breakage caused by heavy ceramic plate caps. During shock, the cap tends to remain stationary relative to the tube. Depending on the direction of the shock motion, the cap may crack the tube envelope or may be unseated entirely, since it is positioned only by friction. A simple metal substitute, as shown in Fig. 58, prevents such damage because of less mass, but the safety hazard is measurably increased. As a compromise, the plastic cap may be used with reasonable success.

Spacing between tubes and adjacent components, in the equipments tested, generally appears to be adequate. Envelope fracture caused by collision can usually be traced to other causes.

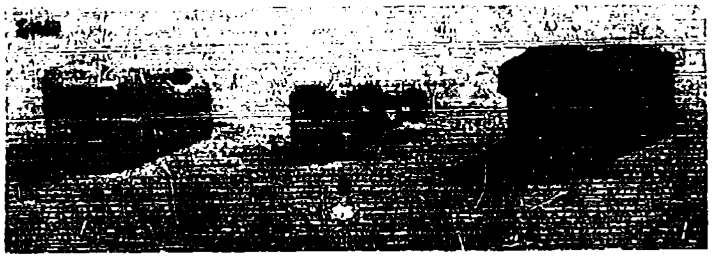


Fig. 58 - Electron plate caps
(a) Ceramic (b) Metallic (c) Plastic

Undoubtedly, further development will remove many present inadequacies in tube design. In the meantime, tube damage can be sharply reduced by proper mounting techniques, good structural design, and the use of ruggedized or reliable versions.

CONCLUDING COMMENTS

Problems of the more common and numerous electronic parts, relating to shock and vibration environments, have been presented. Emphasis has been placed on the necessity of designing the structures of these parts for stiffness and lightness. The same consideration is necessary in the design of less used, or special, parts in order to eliminate sources of malfunctions under shock and vibration. Some concluding considerations are indicated in the following paragraphs.

Ruggedized meters are available for many applications; however, when none is available for particular circumstances, steel-cased meters (nonruggedized), mounted toward the center of a panel, where the greatest deflections occur, stand a better chance of surviving shock than molded phenolic-cased types. Large, stiff rubber grommets, if used to float the meters on the panel, will help to reduce the shock transmission.

Wiring and lead breakages create a major design problem, and damage of this sort is particularly provoking and serious because of the long delays and oftentimes elaborate equipment required to isolate the failure. The use of stranded wire, proper lacing, and clamps for cables and wire bundles, and the inclusion of sufficient slack at terminal connections to accomodate the relative motions occurring during shock and vibration, will greatly simplify the problem.

Fastener damage in equipments whose resonant frequencies are out of testing ranges is minor. Rivets, screws, bolts, and welding are preferred types of fasteners; the various quick-release devices are special-purpose fasteners that carry only light loads, and are not normally desirable. The application of ordinary types of stainless steel bolts and screws for comparatively heavy loads should be carefully considered from a strength viewpoint. These ordinary types (18-8) have a low yield point (about 25,000 psi), and therefore will stretch and loosen under high-impact shock, although they have a high ultimate strength and high ductility. Lockwashers should be used in conjunction with screws and bolts. The star type appears to be inferior to the split-ring lockwasher. Impact loadings cause the surface of soft materials, such as aluminum, to deform under the tips of the star washer, thus resulting in a loose fastener.

Laboratory tests, based on actual field operating conditions, indicate that damages due to shock and vibration are basically mechanical. Consequently, elimination of these damages in new equipments depends on the application of known mechanical principles while the equipment is in the design stages. Damage resistance should be primarily a function of design, with less reliance placed on the shock-isolator.

* * *

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APPENDIX A
Stiffness and Its Effect on Shock Stress-Response

Some examples involving the application of static and dynamic forces to simple beam structures will now be presented. These examples illustrate in part the general statement that, for beams having sections common to engineering uses, the stiffer the beam is the greater its strength, i.e., the greater is the bending moment it can withstand for a given maximum stress in the beam. In these examples, there are two families of beams of simple rectangular configuration. In one set, b (breadth) is held constant while d (depth) is varied, and in the other, d is held constant while b is varied. In contrast, a square hollow beam is included to demonstrate an efficient, weight-saving, stiff design. A further advantage of the square section is equal stiffness in two directions. A round section, of course, would have equal stiffness in all directions.

CANTILEVER BEAMS—STATIC CONDITION

Suppose it is desired to support a concentrated load of 120 lb at the end of a steel cantilever beam whose length is 10 in. and whose natural frequency is required to be 25 cps. The problem is to select a suitable cross section. For practical purposes, the structure can be considered a linear, single-degree-of-freedom system. The effect of the weight of the beam is neglected. The formulae required in the solution of this problem are

$$f_n = \frac{1}{2\pi}\sqrt{\frac{12g}{D}}, \tag{A1}$$

where f_n = natural frequency, $g = 32.2\ \text{ft/sec}^2$ and D = static deflection due to load (inches), and

$$D = \frac{P_s l^3}{3EI}, \tag{A2}$$

which is the deflection at the free end of a cantilever beam, where D = static deflection P_s = static load, l = length of beam, E = modulus of elasticity of material (steel = 30×10^6), and I = moment of inertia of section about its neutral axis.

$$I = \frac{1}{12}\, bd^3 \tag{A3}$$

is the moment of inertia of a rectangular section, where b = breadth and d = depth.

$$S_{max} = \frac{M_{max}\, C}{I} \tag{A4}$$

is the formula for maximum flexural stress of a beam, where M_{max} = maximum bending, C = distance from neutral axis of the beam to outermost fiber, and I = moment of inertia of section about its normal axis.

$$M_{max} = Pl \tag{A5}$$

is the formula for maximum bending moment of a cantilever beam (at fixed end), where P_s = static load and l = length. Solving for D from Eq. (A1),

$$D = \frac{g \times 12}{(2\pi f_n)^2} = \frac{32.2 \times 12}{(6.28 \times 25)^2} = \frac{386}{24,800} = 0.016 \text{ in.}$$

Solving for I from Eq. (A2),

$$I = \frac{120 \times (10)^3}{3 \times 30(10)^6 \times 0.016} = \frac{1}{12} (\text{in.})^4.$$

From Eq. (A3), it can be seen that a section 1-in. square has a value of $I = 1/12$ (in.)^4. From Eq. (A4), the maximum fiber stress at the fixed end of the beam

$$S_{max} = \frac{1200 \times 1/2}{1/12} = 7200 \frac{\text{lb}}{\text{in.}^2}.$$

Table A1 summarizes the results of these calculations, as well as those for beams of other cross sections with l = 10 in. and P_s = 120 lb.

Table A1 shows that in all cases the stiffer beams of a particular family are stronger than the less stiff beams of the same family, since their maximum flexural stresses are always smaller. Another way of considering this is that the beams with higher natural frequencies are stronger than the lower frequency beams for a given loading. The design engineer who has confined his talents to static structures might be tempted to comment that all the beams except beam 1 have been over-designed, assuming the beam material is SAE 1020 cold-rolled steel with a yield of 50,000 psi and a maximum stress of 90,000 psi. Subsequent descriptions and examples of the response of these beams to specific dynamic loadings for shipboard environment will demonstrate that beams 1, 2, and 4 are definitely under-designed. Beam 6 is obviously an efficient design, since ineffective material at the center has been removed. This results in a comparatively lighter and stronger beam, as a glance at the beam weight column of Table A1 will indicate. Figure A1, a plot of maximum static fiber stress versus weight for a 25-cps and 70-cps family of beams, is given to demonstrate that, for a given weight of material, low-frequency structures are initially stressed to a greater degree than high-frequency structures. This fact demonstrates that there is less margin for the added stresses of dynamic conditions, with greater susceptibility to excessive yielding and fracture. This is explained further in the following paragraphs. However, extremes that involve very thin sections which are susceptible to buckling should be avoided. The beams of Fig. A1 are all cantilever beams 10 in. long with a concentrated load of 120 lb at their ends.

CANTILEVER BEAMS—DYNAMIC CONDITIONS

Let us assume that a force P_o is applied to the cantilever structure (beam 2, f_n = 25 cps) in the form of a sinusoidal pulse of t = 1 millisecond (ms) duration. Let us assume further that this force P_o is an inertial loading caused by a sinusoidal acceleration pulse with a peak value of 5 g. This means that the peak dynamic force reaches a value of 5 × 120 = 600 lb. The same inertial effect is realized if a 5-g acceleration pulse is applied to the base of the structure supporting the cantilever beam. This dynamic loading is superimposed on the static loading of 120 lb which the beam is supporting.

TABLE A1
Static Conditions, 10-in. Cantilever Beam

Section	I (in.)^4	D (in.)	f_n (cps)	S_m (psi)	Beam Wt (lb)	K (lb/in)
(1) 1 in. × 1/2 in.	1/96	0.128	8.8	28,800	1.55	938
(2) 1 in. × 1 in.	1/12	0.016	25.0	7,200	3.10	7,500
(3) 1 in. × 2 in.	2/3	0.002	70.0	1,800	6.20	60,000
(4) 1/2 in. × 1 in.	1/24	0.032	17.5	14,400	1.55	3,750
(5) 2 in. × 1 in.	1/6	0.008	35.0	3,600	6.20	15,000
(6) 1-1/2 in. × 1-1/2 in. (1 in. sq)	0.339	0.004	50.0	2,655	3.90	30,770

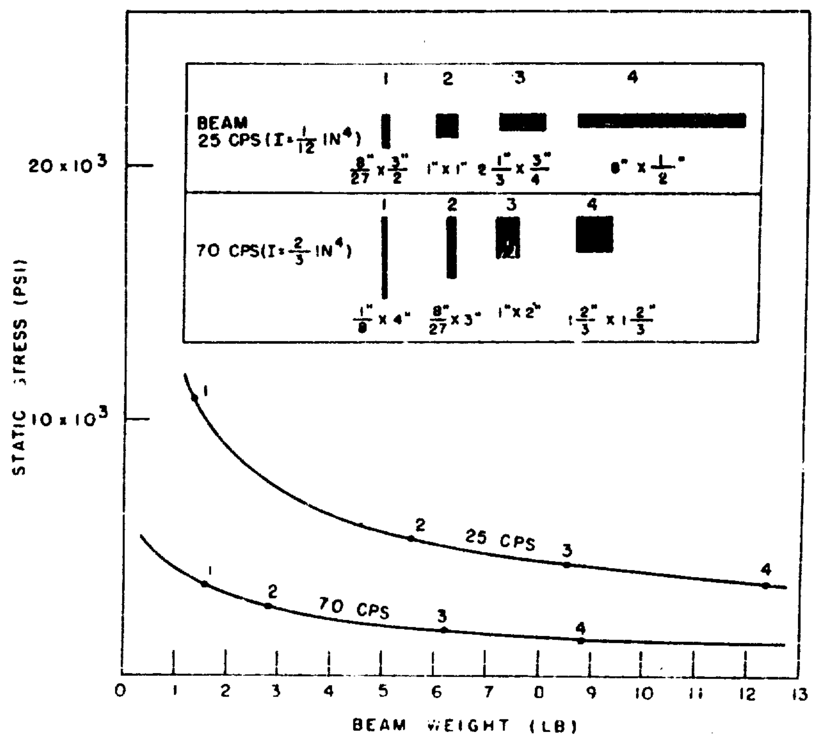


Fig. A1 - Static stress vs. beam weight for 25-cps and 70-cps 10-in. cantilever beams

This level of shock acceleration is representative of that which might be expected in truck transportation, and is not representative of the higher shock levels experienced by naval vessels due to noncontact underwater explosions. This is discussed later. From Table A1, the spring constant of the beam is 7500 lb per in., and the static deflection corresponding to a static load of 600 lb is 0.08 in. Now $t_1/T = 0.001/0.04 = 0.0025$, and, referring to Fig. B3b of Appendix B, the dynamic load factor (DLF) is about 0.10. Therefore, the added loading on the beam due to this force P_0 being applied as a sinusoidal pulse of 1 ms duration is $0.10 \times 600 = 60$ lb; the increase in fiber stress at the location of maximum bending moment is

$$\frac{60 \times 10 \times 1/2}{1/2} = 3600 \text{ psi.}$$

Applying the same pulse to beam 3 ($f_n = 70$ cps) $t_1/T = 0.001/0.014 = 0.071$, and, referring to Fig. B3b of Appendix B, the DLF is about 0.28; the added load due to the dynamic force P_0 applied as a sinusoidal pulse of 1 ms duration is $0.28 \times 600 = 168$ lb, and the increase in fiber stress at the location of maximum bending moment is

$$\frac{168 \times 10 \times 1}{2/3} = 2520 \text{ psi.}$$

Now let us see what happens when the sinusoidal pulse duration is increased to 40 ms, but the peak force-amplitude remains the same, namely 600 lb. For beam 2, $t_1/T = 0.04/0.04 = 1$, and, referring again to Figure B3b, of Appendix B, the DLF becomes 1.75. The added load due to dynamic effects is therefore $1.75 \times 600 = 1050$ lb, and the increase in fiber stress is

$$\frac{(1050 \times 10) \times 1/2}{1/2} = 63{,}000 \text{ psi.}$$

Applying the same pulse ($P_0 = 600$ lb, $t_1 = 40$ ms) to beam 3 ($f_n = 70$ cps), $t_1/T = 0.040/0.014 = 2.9$, the DLF is about 1.2. The added load due to dynamic effects is $1.2 \times 600 = 720$ lb, and the increase in fiber stress is

$$\frac{(720 \times 10) \times 1}{2/3} = 10{,}800 \text{ psi.}$$

In a similar manner, the maximum dynamic flexural stress can be calculated for different sinusoidal pulse durations and for beams of different frequencies. Table A2 summarizes the results of these calculations for dynamic conditions for the 6 beams of Table A1, and the results are plotted in Fig. A2. The total maximum fiber stress in the beams is the sum of the useful static stress due to the load of 120 lb plus the stress due to dynamic loading ($P_0 = 600$ lb) as determined from the DLF. In Fig. A2, the stress at zero pulse duration represents the initial static stress due to the useful load $P_s = 120$ lb. The dynamic stress, due to $P_0 = 600$ lb being applied in the form of a sinusoidal pulse of varying durations up to 40 ms, is added to the static stress. The portions of the curves above the elastic-limit stress (50,000 psi) are dotted to indicate that, above this value, the beam systems are nonlinear, but that the dynamic responses are based on a completely linear system.

As pointed out previously, the intensity of shock is a function of the rate of change and the force magnitude of the shock pulse. Figure A2 also shows that low-frequency structures of reasonable engineering design, with regard to weight considerations, are not only usually initially stressed to a higher degree than high-frequency or stiff structures, but are also more highly stressed as the pulse duration is lengthened, even though the DLF is smaller than for stiff structures. Compare the DLF and stresses of the 8.8- and 70-cps beams of Table A2. It is obvious, therefore, that the low dynamic load factors associated with

TABLE A2
Summarization of Calculations for Dynamic Conditions for the Beams of Table A1

Section	f_n (cps)	T (sec)	t_1 (sec)	t_1/T	DLF	DL	DBM (in-lb)	DBS (psi)	Totals* (psi)
I	8.8	0.11	0.001	0.0091	0.04	24	240	5,760	34,560
			0.010	0.091	0.36	216	2,160	51,840	80,640
			0.020	0.182	0.72	432	4,320	103,680	132,480
			0.030	0.273	1.10	660	6,600	158,400	187,200
			0.040	0.364	1.30	780	7,800	187,200	216,000
II	25.0	0.040	0.001	0.025	0.10	60	600	3,600	10,800
			0.010	0.250	1.00	600	6,000	36,000	43,200
			0.020	0.500	1.60	960	9,600	57,600	64,800
			0.030	0.750	1.75	1,050	10,500	63,000	70,200
			0.040	1.000	1.75	1,050	10,500	63,000	70,200
III	70.0	0.014	0.001	0.071	0.28	168	1,680	2,520	4,320
			0.010	0.710	1.73	1,038	10,380	15,570	17,370
			0.020	1.420	1.50	900	9,000	13,500	15,300
			0.030	2.130	1.20	720	7,200	10,800	12,600
			0.040	2.840	1.20	720	7,200	10,800	12,600
IV	17.5	0.057	0.001	0.0175	0.1	60	600	7,200	21,600
			0.010	0.1750	0.7	420	4,200	50,400	64,800
			0.020	0.3500	1.2	720	7,200	86,400	100,800
			0.030	0.5250	1.6	960	9,600	115,200	129,600
			0.040	0.7000	1.75	1,050	10,500	126,000	140,400
V	35.3	0.0284	0.001	0.0352	0.14	84	840	2,520	6,120
			0.010	0.352	1.35	810	8,100	24,300	27,900
			0.020	0.704	1.74	1,044	10,440	31,320	34,920
			0.030	1.056	1.73	1,038	10,380	31,140	34,740
			0.040	1.408	1.50	900	9,000	27,000	30,600
VI	50.0	0.02	0.001	0.05	0.20	120	1,200	2,655	5,310
			0.010	0.50	1.60	960	9,600	21,286	23,941
			0.020	1.00	1.75	1,050	10,500	23,205	25,860
			0.030	1.50	1.50	900	9,000	19,890	22,545
			0.040	2.00	1.30	780	7,800	17,238	19,893

*These totals represent the values of the dynamic bending stresses (DBS) plus the values of the static stresses (S_m) shown in Table A1.

low-frequency structures under shock will not necessarily assure an acceptable stress level. The acceleration-pulse durations associated with Navy high-impact shock machines are of the order of 1-2 milliseconds, and these durations are reasonably close to those determined in underwater explosion field tests. However, if for some reason these values were exceeded, the problem would become more acute for the low-frequency structures. This fact is also shown in Fig. A2 by the initial slope of the dynamic-stress response curves as well as by the maximum-stress response.

CANTILEVER BEAMS—DYNAMIC CONDITIONS, SHIPBOARD

Figure A2 indicates the maximum stress response of the beams to a sinusoidal pulse of 5 g's magnitude, and, for this value, up to pulse durations of 5 milliseconds, they are

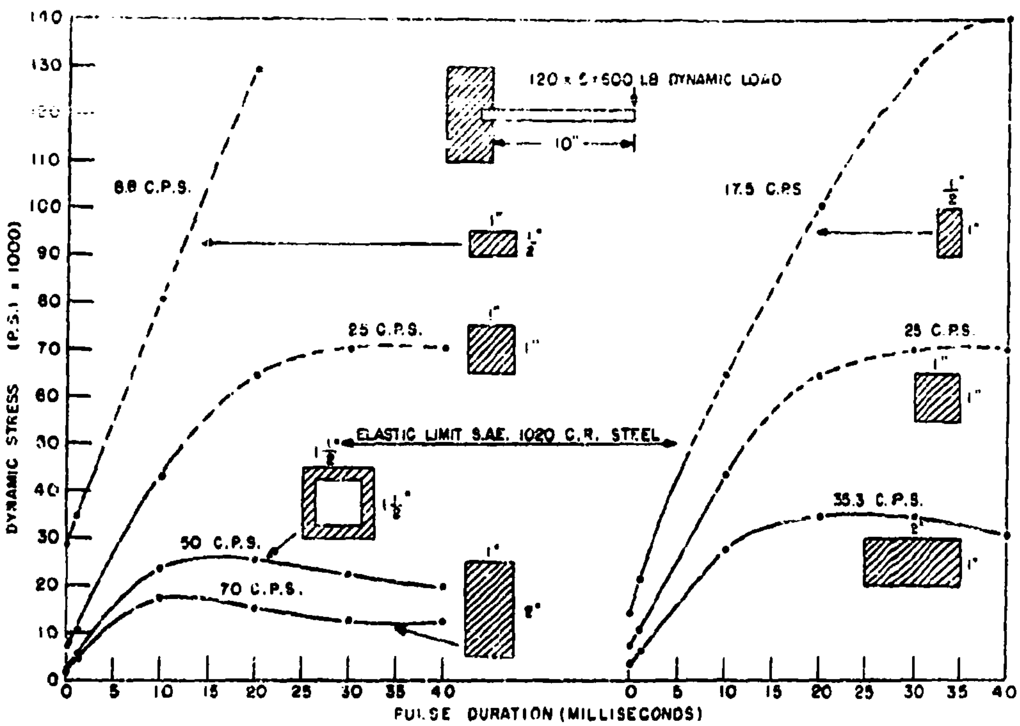


Fig. A2 - Dynamic response for beams of Table A1 to a sinusoidal acceleration pulse of 5 g

acceptable. However, this magnitude is what may be expected in electronic equipment installed in trucks or trailers. For shipboard conditions, the values of accelerations are higher, up to 100 g's or more for the rigid body motion or motion of the center of mass of the equipment. If the magnitude of the pulse is increased to 50 g's, the stress values would then be 10 times greater. For pulse durations of 2 milliseconds, the maximum stresses for the beams identified by frequency would be as follows; 8.8 cps - 40,000 psi, 25 cps - 150,000 psi, 50 cps - 80,000 psi, 70 cps - 65,000 psi, 17.5 cps - 280,000 psi, 35.2 cps - 85,000 psi. It is concluded from these stress values that the 8.8-cps, 17.5-cps, and 25-cps beams would fracture under a 50-g shock of 2 milliseconds duration. The 35.3-cps and 50-cps beams would show visual evidence of plastic deformation, while the 70-cps beam's plastic deformation would be much smaller, and perhaps require instruments to measure.

FIXED END BEAMS—STATIC CONDITION

Actually, a cantilever-type structure does not permit full utilization of the energy-absorption capacity of the material. Furthermore, the utilization of such a structure in electronic equipment (except for antennas) is highly improbable, and more than likely a beam fixed at both ends would be utilized. Now, consider the problem to be identical in all respects to the cantilever beam, but that the beams are fixed at both ends, and the concentrated load of 120 lb is applied at the center. The length of the span is still 10 in., but the beam itself will now be 12 in. long instead of 11 in., as in the case of the

TABLE A3
Static Conditions, 10-in. Fixed-End Beam

Section	I (in.)^4	D (in.)	f_n (cps)	S_m (psi)	Beam Wt (lb)	K (lb/in)
(1) 1 in. × 1/2 in.	1/96	0.002000	70	3,600	1.69	60,000
(2) 1 in. × 1 in.	1/12	0.000250	200	900	3.38	480,000
(3) 1 in. × 2 in.	2/3	0.000031	560	250	6.76	3,840,000
(4) 1/2 in. × 1 in.	1/24	0.000500	140	1,800	1.69	240,000
(5) 2 in. × 1 in.	1/6	0.000125	280	450	6.76	960,000
(6) 1-1/2 in. × 1-1/2 in., 1 in. sq	0.339	0.0000625	400	333	4.24	1,969,280

cantilever beams. The extra inch of length is the allowance for fixing the other end of the beam. The static deflection of a beam fixed at both ends with a concentrated load at its center is

$$D = \frac{P_s l^3}{192EI}. \tag{A6}$$

It is apparent that the static deflection of the fixed-end beams of the same cross section and material as the cantilever beams is 64 times less, and the natural frequencies are 8 times greater. The maximum bending moment occurs equally at the center and ends, and has a value given by

$$M_{max} = \frac{P_s l}{8}. \tag{A7}$$

From Eq. (A7), it can be seen that, since the maximum bending moment is 8 times less than the maximum bending moment for the cantilever beam, the maximum stresses as determined from Eq. (A4) are 8 times less. However, this double fixed-end beam is strained to a greater degree over its length for equal deflections, and therefore has a greater energy-absorption capacity per unit of weight or volume than the cantilever beam. This is the essence of efficient design. Table A3 summarizes the results for static

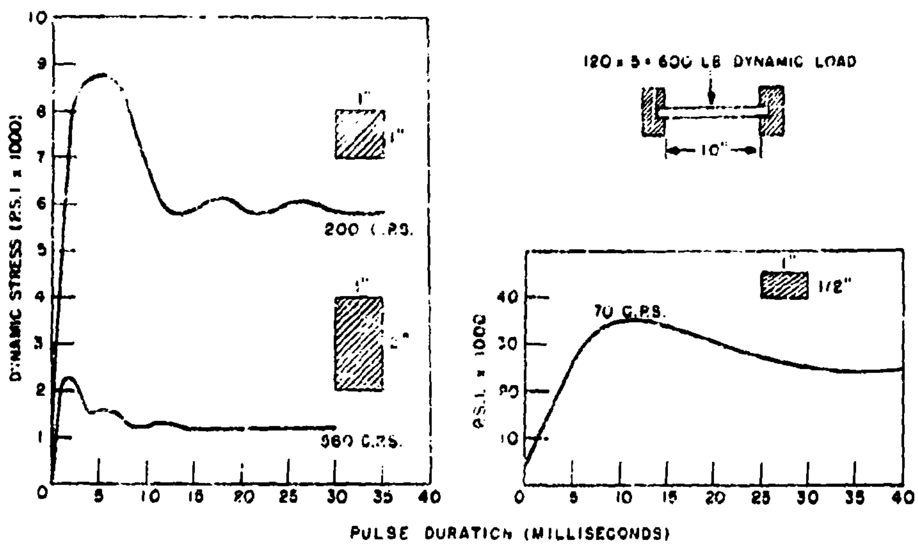


Fig. A3 - Dynamic response for beams of Table A2 to a sinusoidal acceleration pulse of 5 g

conditions for the 10-in. beams fixed at both ends with a concentrated load of 120 lb at their centers. The cross sections and materials are the same as in Table A1.

FIXED-END BEAMS—DYNAMIC CONDITION

Figure A3 shows the results for three fixed-end beams under the same dynamic conditions as for the cantilever beams. The cross sections shown are (1), (2), and (3) of Table A3. All these beams are satisfactory for the 5-g acceleration pulse, regardless of pulse durations. Referring back to Fig. A2 for the stress response of the cantilever beams, this conclusion is not true for the 8.8- and 17.5-cps beams while the 25-cps beam is marginal. However, this conclusion is true for the 35.3-, 50-, and 70-cps cantilever beams.

FIXED-END BEAMS—DYNAMIC CONDITIONS, SHIPBOARD

If the pulse magnitude is increased to 50 g's, a shock level more representative of shipboard environment, the stress values would be 10 times greater, and for a 2-millisecond pulse duration, the maximum stresses for the beams identified by frequency would be as follows: 70 cps - 130,000 psi, 200 cps - 79,000 psi, and 560 cps - 23,000 psi. The latter beam, it should be emphasized, would be considered satisfactory for shock levels in excess of 100 g's, regardless of pulse duration. Moreover, beam-section (2), which is 1-in. square, when used for the cantilever, results in a 25-cps beam with a dynamic load factor of about 0.2 for a 2-ms pulse duration. When used for the double fixed-end beam, it produces a 200-cps beam with a dynamic load factor of about 1.11 for a 2-ms pulse duration. The stresses, however, for this dynamic condition, are 150,000 psi and 79,000 psi, respectively. This emphasizes the fact that a low value of the dynamic load factor may be misleading when not correlated with the associated stresses of the structural sections involved. This does not mean that acceptable stress levels cannot be achieved with low-frequency structures, but the required additional material to accomplish this results in increased weight, as shown in Fig. A1.

FIXED-END BEAMS—DISTRIBUTED LOAD

If the same load is distributed, rather than concentrated, the ability of the beams to resist dynamic loading is still further increased. For the same beams, fixed at both ends with a uniformly distributed load, the maximum deflection is given by

$$D = \frac{wl^4}{384EI}, \tag{A8}$$

where w is the weight per unit length. Comparing Eqs. (A6) and (A8), it is seen that the uniformly loaded beam has a static deflection for the same load equal to one-half of the value for the beam with the same concentrated load, and therefore the natural frequency is increased further by the factor $\sqrt{2}$. The maximum bending moment occurs at the ends, and has a value given by the following equation:

$$M_{max} = \frac{wl^2}{12}. \tag{A9}$$

Comparing Eqs. (A7) and (A9) it is seen that, for identical conditions, the maximum flexural stresses for the uniformly loaded beams, as determined by Eq. (A4), would be 1.5 times less.

This analysis of the dynamic response of beams indicates the basic considerations in designing for high-impact shock. The analysis of more complicated structures, such as chassis and columns, is a difficult problem, and more investigation is required. However, the practical aspects of the problems involved have been thoroughly explored in the long-term evaluation program of electronic equipment, and the suggested design methods and considerations presented in this report have contributed toward the elimination of many failures and attainment of reliable equipment. Compliance with these practical guides will eliminate most of the unnecessary or avoidable difficulties which are repeated so many times in evaluation procedures.

Other practical considerations, such as degree of end-fixity of the beams, and eccentric loading (producing torsion), may be very important factors in the actual performance of the beam. The glib assumption of a fixed-end beam may not be so easy to attain in the actual equipment and, if overlooked, a modification during testing might be necessary to achieve this, by the introduction of a doubler, or stiffener, or gusset connections to the side panel of the equipment.

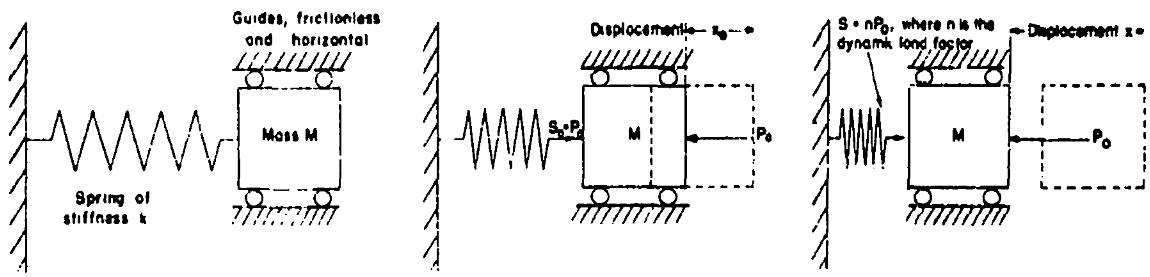
* * *

APPENDIX B
Dynamic Load Factor

This material on the dynamic response of single-degree-of-freedom systems to simple shock motions is taken from the David W. Taylor Model Basin Report 481, entitled "Effects of Impact on Simple Elastic Structures" by J. M. Frankland.

All structures possess many degrees of freedom, but frequently one is so preponderant as to determine the behavior of the system, for all practical purposes. A structure is considered here which possesses a single degree of freedom; it is exemplified by a rigid mass, supported without friction or dam, attached to an inertialess spring, as shown in Fig. B1a.

Figure B1b illustrates the case of static loading on this system, in which the force P_0 causes a displacement of the mass M and a shortening of the spring by the amount x_0. When the application of the force or load P is a function of time, as in Fig. B1c, dynamic loading occurs; the displacement x and the spring force S can then be conveniently related to the corresponding values for static loading by the use of certain nondimensional ratios.



(a) System having one degree of freedom

(b) System showing static displacement x_0 of the mass by a steady force P_0

(c) System showing dynamic displacement x of the mass by a disturbing force P whose peak value is P_0

Fig. B1 - Undamped elastic system

The ratio between the applied force P at any moment during the dynamic loading period and the maximum value of the force P_0 applied to the system is designated as the disturbance factor, or, more briefly, the disturbance. As P never exceeds P_0, the maximum value of this factor is 1.

The static displacement x_0 of the mass M under the steady load P_0 can, as shown in Fig. B1b, be used correspondingly as a unit of displacement. Under sudden application of the force, P, the displacement rises to a dynamic value x, as in Fig. B1c; the ratio of this dynamic displacement x to the static displacement x_0 is called the response factor, or, more briefly, the response. As shown by a comparison of Figs. B1b and B1c, the maximum value of this factor may greatly exceed 1.

As the spring reaction S is assumed to follow Hooke's law, the response factor may be used to represent ratios of spring force, or load, as well as of displacement or deformation. Thus, the reactive force S exerted by the spring at any time is the maximum

force P_0 multiplied by the response factor at that instant. The numerical maximum of the response factor, derived from the ratios x/x_0 or S/S_0, is the dynamic load factor; it is the factor which, multiplied into the maximum load P_0, gives the maximum spring reactive force under the dynamic condition defined by the disturbance. Whereas S_0 always equals P_0 in static loading, the reactive force S exerted on and by the spring, or equivalent supporting structure, may greatly exceed the instantaneous value of the applied load P under dynamic loading conditions.

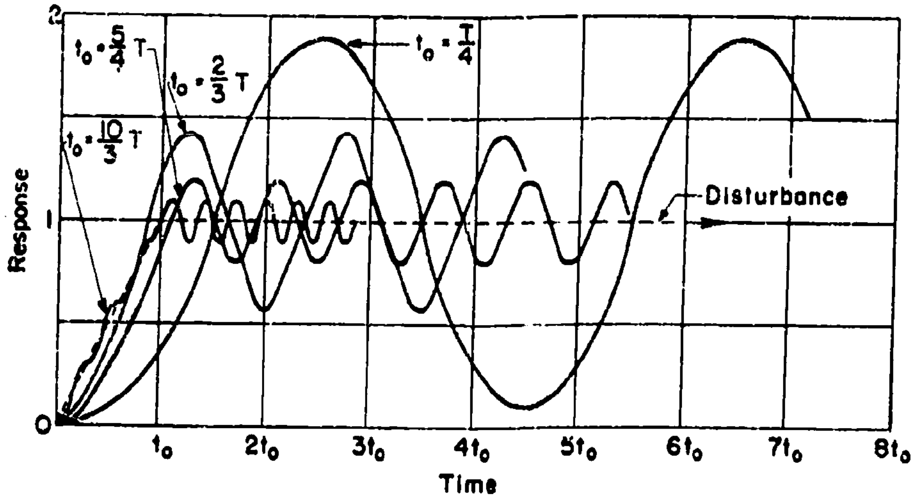
The factor 2, to be found in all text books on mechanics for determining the force equivalent of a suddenly applied load, is a dynamic load factor. Knowledge of the dynamic load factor is a prerequisite to the design of a structure to resist a particular shock load.

If the maximum load on the spring is such that the elastic range of the material is not exceeded, the stress in the spring will be at all times proportional to the reactive force S. Since it has previously been shown that the ratio S/S_0 is equal to the ratio x/x_0, the response factor, expressed by the latter ratio, can also be used as a ratio of stresses in the spring material. Since the spring shown in Fig. B1 can be replaced by an equivalent elastic structure having one degree of freedom, such as a long, slender beam, it can be said that the maximum value of the response factor, which is equal to the dynamic load factor, gives the ratio of maximum stress in the beam under the shock or impact load to the stress set up in the beam by a static load P_0 equal to the peak load P in the shock pulse, i. e., under the shock loading conditions.

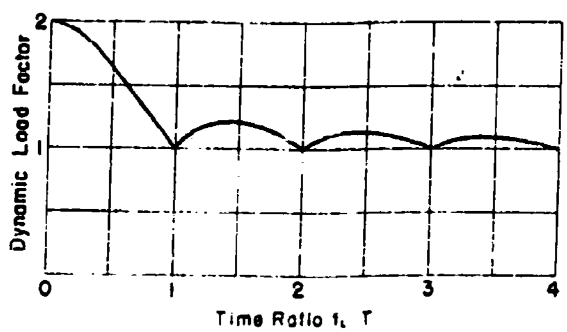
Calculation of the response factor for several types of disturbance shows that the rate of application of the load and the duration of the load are of primary importance in a determination of the stresses set up in any particular structure. If the load is instantaneously applied, and remains constant for a duration exceeding half the natural period T of vibration of the structure, the dynamic factor is 2; that is, stresses are double those obtained on applying this load longer than is required to give the mass M in Fig. B1 a displacement of x_0, or to push it to a greater displacement x, and have it return to its initial position as shown in Fig. B1a, as it will do under the influence of a maximum reactive force S much greater than P. If the duration t_1 falls below T/2, then in all cases the dynamic load factor decreases from 2 to 0 as the ratio t_1/T drops to zero.

If the load is not instantaneously applied, and if the time of rise t_0 to peak load is less than one fourth the natural period T of the structure (Fig. B2), then very nearly the maximum effect due to rate of application of load is realized; this is because the mass M in Fig. B1 is travelling to the left for more than the entire duration of the increasing push exerted by the load. As t_0 becomes larger than T/4, the dynamic load factor progressively decreases.

For a disturbance similar to the first half-cycle of a sine wave (Fig. B3), the largest dynamic load factor is for a duration t_1 of about one natural period, where the increase in stresses over the corresponding case of static loading is about 75 percent. [illegible] the type of disturbance caused by gun blast (Fig. B4), [illegible] it be assumed that the load rises instantaneously to its maximum, the dynamic load factor begins to decrease markedly if the duration of positive pressure falls below about 4T. However, if the rise is not instantaneous (Fig. B5), increasing to its maximum in the time t_0, then the dynamic load factor decreases at a rate determined by the ratios t_0/T and t_1/T.

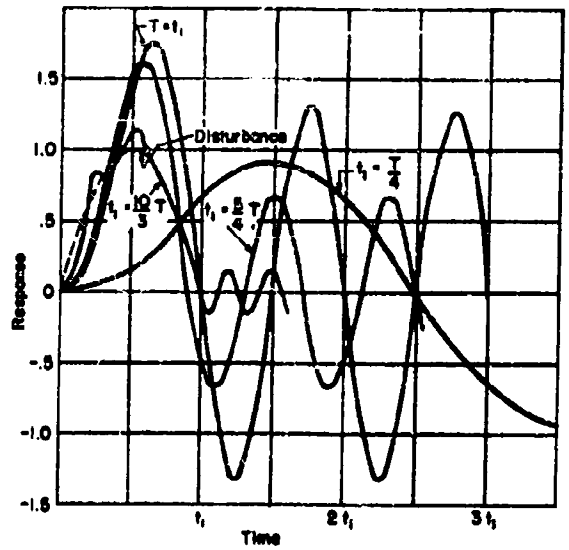


(a) Response of various systems to gradually applied loads

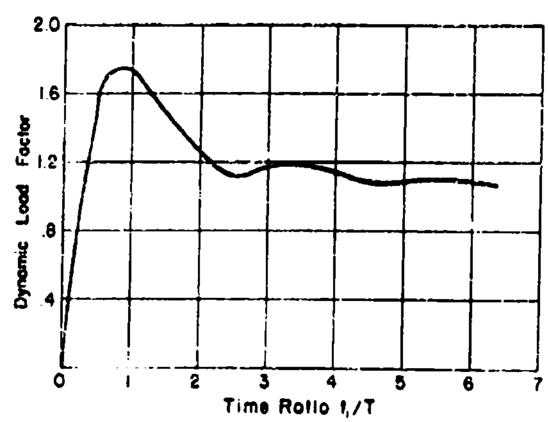


(b) Dynamic load factor for the disturbance of the type shown in Fig. B2a

Fig. B2 - Disturbance applied gradually and maintained indefinitely

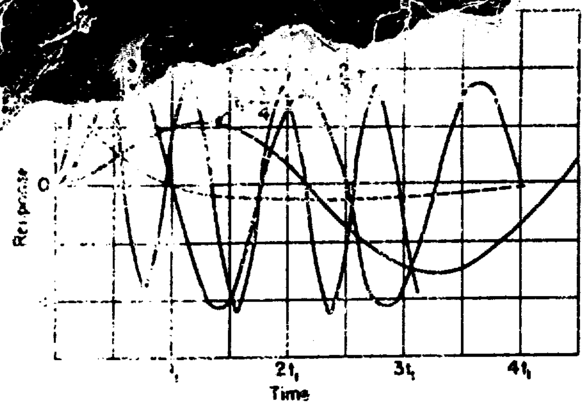


(a) Response to a sinusoidal pulse

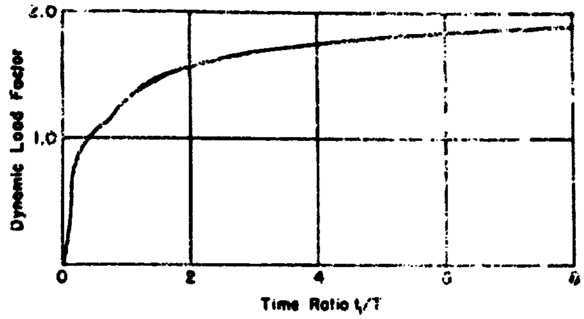


(b) Dynamic load factor for a sinusoidal pulse

Fig. B3 - Sinusoidal pulse

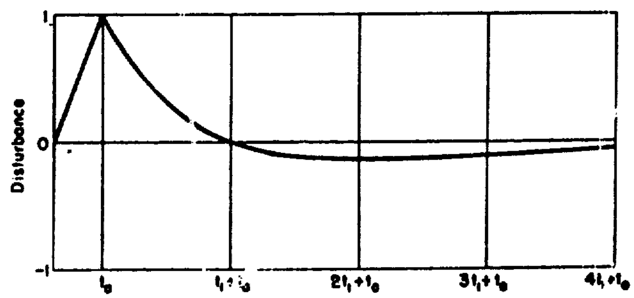


(a) Response to blast pulse. Disturbance is shown by broken line.

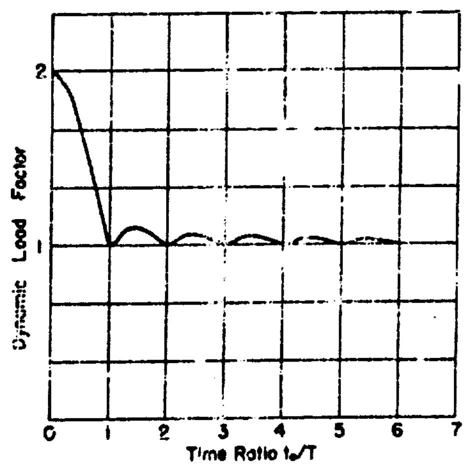


(b) Dynamic load factor for blast disturbance of the type shown in Fig. B4a

Fig. B4 - Blast pulse



(a) Plot of modified blast disturbance



(b) Dynamic load factor for blast disturbance of the type shown in Fig. B5a

Fig. B5 - Modified blast pulse

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